FOULING ANALYSIS AND ITS MITIGATION IN HEAT EXCHANGERS

BY

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DISSERTATION ABSTRACT

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The fouling growth has been studied using thermal, second law and statistical analyses for two heat exchangers in oil industry. Moreover, the effectiveness of two new fouling mitigation technologies, namely helically baffled heat exchanger and self-cleaning heat exchanger, are evaluated. The results indicate that the helically baffled heat exchanger is very effective in reducing the shell side fouling and the self-cleaning heat exchanger is very effective in eliminating the tube side fouling. For the thermal analysis, comparison between the Bell Delaware and Flow Stream Analysis methods indicate that the Bell Delaware method gives a relatively more accurate fouling growth results. The evaluations based on second law of thermodynamics show that the irreversibility is increasing with the increase of fouling growth for the exchanger with the same thermal and hydraulic inlet conditions. This finding could be used by plants' engineers for easy fouling monitoring. The fouling growths from the statistical models were obtained and compared with the fouling growth obtained from the thermal analysis. The comparison indicates that there is no much difference in the fouling growth obtained from the two methods. Therefore, the statistical method represents an easy and accurate approach. The results of the study are presented in tabular and graphical forms and discussed in details.

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تمت دراسة زيادة الترسبات في المبادلات الحرارية باستخدام التحليل الحراري، القانون الثاني الديناميكي الحراري والتحليل الاحصائي. وايضا لقد تمت دراسة كفاءة تقنيتين جديدتين لإزالة الترسبات في المبادلات الحرارية. هذه التقنيتين هما المبادلة الحرارية ذات الدعائم الانسيابية والمبادلة الحرارية ذات التنظيف الذاتي. ولقد اظهرت النتائج فعالية هاتين التقنيتين في إزالة الترسبات من المبادلات الحرارية في جهة الانبوب وفي جهة الغلاف للمبادلة الحرارية. بالنسبة للتحليل الحراري، تمت المقارنة بين طريقة بل ديلوير وطريقة سيل التدفق وقد اظهرت النتائج ان طريقة بل ديلوير تعطي نتائج ادق. وقد أظهرت النتائج باستخدام القانون الثاني للديناميكا الحرارية أن الانعكاسية تزداد بزيادة الترسبات للمبادلة التي لايتغير فيها الحالة الحرارية والهيدروليكية للسوائل الدلخلة. بإستخدام نتائج هذا البحث، يستطيع المهندسون في المعامل المتابعة السهلة للترسبات في المبادلات الحرارية. وايضا تم الحصول علي الترسبات باستخدام التحليل الاحصائي وتمت مقارنة النتائج مع نتائج الترسبات من التحليل الحراري. وقد اظهرت النتائج انه لافرق بدقة النتائج بين الطريقتين. ولذلك يعتبر التحليل الاحصائي طريقة سهلة ودقيقة لمتابعة الترسبات. وقد تم عرض النتائج في جداول ورسوم توضيحية كما تم مناقشتها بالتفصيل.

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CHAPTER 1

INTRODUCTION

Fouling is the major unresolved problem in heat transfer equipment [1]. It is defined as the deposition of unwanted by-product on the heat exchanger surface that reduces heat transfer capability and increases flow resistance. It is an unavoidable by-product of heat transfer process [2, 3, 4].

During operation, heat exchanger surfaces can become fouled to a greater or lesser extent [5]. The growth of these fluid impurities causes the thermal-hydraulic performance of heat transfer equipment to decrease with time. In addition, where the heat flux is relatively high, as in steam generators, fouling can lead to local hot spots and ultimately it may result in mechanical failure of heat transfer equipment and hence it results in an unscheduled shut down of the plant [6].

Fouling has been a very serious problem in heat exchangers ever since their invention. Fouling increases the resistance for both heat transfer and fluid flow. In order to make up for these negative effects, initial size of the heat exchanger has to be increased to account for this fouling effect. An increase of up to 30% has to be added to the nominal size to compensate for the losses in heat transfer. This increase adds not only to the initial cost of an exchanger but also to the operational costs. High operational costs are incurred including: periodic cleaning, extra energy, additions of expensive anti-scalant and anti-oxidant chemicals, shut down cost, and lost production as a result of reduced on-stream factors [7]. Curlett and Impagliazzo [8] used the main condenser of a large power plant as an example to demonstrate the consequence of heat exchanger tube fouling. They demonstrated the effect of fouling on the plant performance and cost. Also, they presented a technique for economical evaluation of fouling. They reported that actual predication of fouling can minimize cost of generated power. There is no accurate figure about the cost of fouling, but, there is no doubt about its high cost and its impact on the economy. Muller-Steinhagen [9] states "in year 1993, the costs caused by the process related formation of deposits on heat transfer surfaces are in the order of 40,000 million US dollars per year for the total industrial world". He also estimated that fouling costs for highly industrialized countries such as the USA and UK are about 0.25% of the countries' Gross National Product (GNP).

In this study, the fouling growth characteristic in two shell and tube heat exchangers, which are subjected to severe fouling, are investigated. In addition to the traditional thermal analysis approach, a statistical method is also proposed to predict fouling at any time. For minimizing fouling, two new fouling mitigation technologies are presented and their effectiveness in mitigating or minimizing fouling are evaluated. The two heat exchangers under consideration (as examples) are naphtha heat exchanger (exchanger case 1) and thermosyphon reboiler (exchanger case 2).

A detailed literature review about the fouling analyses and some of the current mitigation technologies are presented in chapter 2. Also, the objectives of this study are introduced at the end of chapter 2.

The problem description for both the naphtha exchanger (exchanger case 1) and the thermosyphon reboiler (exchanger case 2) is presented in chapter 3. As examples, the fouling analysis and the mitigation techniques are used and tested on these two exchangers, which have a history of severe fouling. Exchanger case 1 has shell side fouling problem while exchanger case 2 has tube side fouling problem.

Chapter 4 describes the fouling calculations based on thermal analyses. Moreover, the on-line experiments for both exchangers are introduced. Then, the solution procedure and the error estimation in calculating fouling are discussed. Finally, the results and discussions for the two heat exchangers are presented in this chapter.

Chapter 5 is about fouling prediction. This chapter introduces the statistical approach, which is used to develop fouling growth models. This approach is applied on both exchanger case 1 and exchanger case 2. The fouling models that are discussed in this chapter could be used to schedule the maintenance of the exchangers.

In chapter 6, using the second law of thermodynamics and heat exchanger thermal analysis, a relationship between the irreversibly and fouling resistance is developed. The effect of fouling on the irreversibility for exchanger case 2 is discussed in this chapter. Similar to the heat exchanger cleanness factor (U/U_c), the heat exchanger performance factor (I_c/I) is presented for exchanger case 2 as a function of fouling resistance.

Chapter 7 introduces two fouling mitigation technologies namely helically baffles to mitigate shell-side fouling and self-cleaning to mitigate tube-side fouling. Also, the effectiveness of these new technologies to mitigate the fouling problems in both exchanger case 1 and exchanger case 2 is discussed.

Chapter 8 summarizes the outcome of this research and suggests some future work that could be carried out as a continuation of this work.

CHAPTER 2

LITERATURE REVIEW

In view of the fact that fouling is the major unresolved problem in heat exchangers since their invention, several studies have been conducted on heat exchangers fouling. Furthermore, many techniques to reduce fouling have been introduced and evaluated.

Many researchers, such as Somerscales [10], Hewitt et al. [5], and Zubair et al. [6], categorized thermal fouling into six types based on dominant cause of fouling. These are precipitation fouling, particulate fouling, chemical reaction fouling, corrosion fouling, biological fouling, and freezing fouling. It should be noted the first five types are promoted by heating and do not involve surface crystallization. Characklis et al. [11] presented a model for simulating a biofouling development and its influence on heat transfer resistance. Somerscales [12] presented the in situ corrosion fouling of non-ferrous heat transfer surfaces exposed to flowing water. Novak [13] studied the fouling rate of two natural waters. He reported that chlorination is one of the most common biofouling control methods and it may be used for removal of already formed microbiological deposits. Ritter [14] studied aqueous solutions circulated through electrically heated tubes at controlled conditions. He found out that the induction periods and fouling rates are primary function of the supersaturation of the solutions. Beal and Armstrong [15] developed a model to predict the particle size distribution in a system with corrosion fouling. Baier et al. [16] reported the experimental results for a program seeking to minimize biological fouling of heat exchanger surface by understanding, predicting, and controlling the earliest event at the material/fluid interface. Duffuaa and Budair [17] presented a method for optimal schedule for scale removal in heat exchange equipment.

Crittenden et al. [18] reported that fouling mechanism can be divided into five steps. Namely, initiation, transport, attachment to the surface, removal from the surface, and ageing. Panchal et al. [19] carried out an investigation to understand the mechanisms for the formation of iron sulfide and polymeric deposition on heat exchanger's surface. They highlighted that the threshold fouling temperature can be linked to the decomposition temperature of iron salts of naphthenic acids.

Several studies have been conducted to investigate and analyze heat exchangers fouling [20, 21, 22]. Xu et al. [23] investigated particulate fouling mechanisms. They developed a new predictive model for particulate fouling. Panchal et al. [24] utilized laboratory and filed data of crude oil fouling to develop a correlation covering a wide range of physical parameters. The new correlation predicts the rate of fouling as well as threshold fouling conditions for a majority of data points. Thomas [25] presented numerical techniques to perform thermal and fouling evaluations for heat exchangers. Shah and Sekulic

[26] presented a step-by-step procedure to calculate the non-uniform overall heat transfer coefficient for conventional heat exchangers with variable thermal-physical properties. This will give more accurate results for fouling in case of variable thermodynamics properties. Newson et al. [27] concluded that the deposition rate of suspensions varies with velocity, temperature and concentration. Joshi [28] presented five fouling situations in a petroleum refinery, varying from cold crude to a light end reboilers. For each case, the fouling analysis and possible solutions were presented. Sadrameli et al. [29] simulated the heavy fouling and operating performance of transfer line exchangers. The simulation predicted the coke formation rate in addition to the physical condensation of heavy hydrocarbons. The results indicated a good match with full-scale plant data. Jones and Bott [30] reported that designing to capacity process yield significant cost savings, faster execution schedules and reduced fouling. Obot et al. [31] carried out experimental investigation to determine the pressure drop and heat transfer characteristics for spiral tubes. The results indicate that there is a definite connection between fouling transition period and the heat transfer enhancement. Arman and Rabas [32] presented a numerical modeling that can be used to predict the pressure drop in heat exchanger tubes with transverse, repeated ribs. The model provides predication for factors that affect fouling rate. These factors include wall shear stress, pressure distribution and velocity distribution. Knudsen et al. [33] determined, based on tests they performed, the threshold fouling curve for a crude oil. Asomaning and Watkinson [34] studied fouling of mixtures of heavy oil. They

reported that fouling is related to the presence of suspended asphaltene entities, the concentrations of heavy oil, and temperature. Brons and Rudy [35] studied the relationships between the whole crude oil properties and fouling. They concluded that the whole crude oil properties are the most responsible fouling and thermal cracking tendencies.

Statistical analysis, which has been used by several researchers, is a good tool to evaluate heat exchanger fouling and the maintenance strategy of fouling. Sheikh and Al-Bagawi [36] performed statistical analyses to characterize the time between cleaning of thermosyphon reboilers in oil industry. They developed very accurate models that represent the reboilers fouling. Also, Zubair et al. [6] developed a maintenance strategy for heat transfer equipment subject to fouling. Sheikh et al. [37], on the other hand, used reliability based maintenance strategies for heat exchangers subject to fouling. Duffuaa and Budair [38] proposed the use of a function, which shows explicitly the utilization of energy over a period of time and a specified level of energy utilization to deduce a schedule for scale removal. Also, they [39] show the impact of scale build up on energy utilization in heat exchangers. Yeap et al. [40] developed an algorithm for simulating fouling behavior in shell and tube heat exchangers. They reported that the program helps to understand the interactions between temperature effects, fluid dynamics and fouling. Bott and Jones [41] described the existing fouling monitoring models. They highlighted that there is a need for a more comprehensive approach and improved methods of fouling measurement and predication. Wilson and Vassiliadis [42] presented a management planning approach to clean heat exchangers in preheat trains. Pugh et al. [43] discussed the development of the user guide, developed by ESDU International. Also they utilized the user guide to evaluate fouling in the preheat train and to provide various practical methods to mitigate and control fouling. Wang et al. [44] proposed an on-line monitoring model of ash deposits on economizer. The model was based on easy measurable parameters such as temperatures and pressure drop. Knudsen [45] provided a brief review of cooling water fouling. He presented different scale models. Also, he discussed the criteria for predicting scale-forming tendency. Leach and Factor [46] used computer based monitoring programs to indicate the relative effectiveness of various solutions to fouling problems in refinery and petrochemical plants.

Moreover, the second law based evaluation of heat exchangers fouling was utilized successfully by many researchers. For example, Yang et al. [47] studied the effects of fouling on the irreversibility of heat exchangers. He concluded that the irreversibility increases with the increase of fouling.

Generally, the conventional methods that are used to mitigate fouling include chemical cleaning and mechanical cleaning. These methods are expensive and require shutting down the heat exchanger. One of these methods is sootblowing. Allmon et al. [48] conducted a unique test to evaluate flue gas fouling and cleaning performance characteristics of a heat exchanger bundle for applications in coal-fired environment. They reported that ash deposits were controlled by sootblowing.

Tube enhancements increase heat transfer and reduce fouling tendency. Somerscales et al. [49] studied the effect of fouling on heat transfer tubes that are enhanced on their inner surface. They reported that enhanced surfaces exhibit lower fouling rates than plain surfaces under the same conditions. Watkinson [50] examined the interactions between heat transfer enhancement and fouling. He reported that the enhancement reduces fouling, increases heat transfer, lowers wall temperature and increases pressure drop. Kim [51] studied the effect of spiral turbulators on shell and tube heat exchangers performance. Moreover, he presented several fouling removal methods. Based on the experimental investigations, he provided relations among the heat transfer, drag, effectiveness of cleaning the foulant, and geometrical variables of cleaning/augmenter. Bott and Bemrose [52] carried out fouling tests for gas-side of finned tubes. They reported the results for 4-row, 4-pass spiral wound finned tube heat exchanger. Xu et al. [53] investigated the fouling characteristics of corrugated tubes. They concluded that the corrugated tube has an advantage over the plain tube in the anti-fouling ability.

Recently, several technologies have been introduced to mitigate heat exchangers fouling. These technologies could prevent both the shell and tube side fouling. Among such technologies are self-cleaning, helical baffles configuration, and ball cleaning.

Self-cleaning technology was developed and evaluated for different applications by Klaren [54, 55] and Klaren and Sullivan [56, 57, 58, 59].

Moreover, Gibbs [60] highlighted that the fluidized bed exchanger eliminates the fouling in a reboiler which has to be cleaned frequently as a result of severe and frequent fouling. Gibbs [61] found out that the heat transfer coefficient doubles with fluidized bed heat exchangers. In this new technology, small particles will be charged with the crude to provide scouring action and clean the tubes continuously. Moreover, it will increase the efficiency of the heat exchanger since these particles will increase the flow turbulence level.

The helically baffled heat exchanger is also a new technology that uses helical baffle configuration instead of the conventional segmental configurations [62]. Helical baffle configuration reduces the shell side pressure drop, fouling and flow-induced vibration and also it enhances heat transfer. Lutcha and Nemcansky [63] investigated the performance of helically baffled heat exchanger. Their investigation revealed that there is a potential improvement in the exchanger performance with the helical baffle configuration. Stehlik et al. [64] compared the heat transfer and pressure drop correction factors based on Bell Delaware method for both the segmental baffled exchanger and the helically baffled exchanger. In general, the results showed that properly designed helical baffles offer a significant improvement in heat transfer while providing a reduced exchanger pressure drop. The results for different baffle configurations was presented and discussed by Keal et al. [65].

Another fouling mitigation that has been used in sea-water heat exchangers is the ball cleaning technology [66]. The ball cleaning system technology utilizes sponge balls to continuously clean the tube side of the exchangers utilizing sea-water as a cooling media. This system could be connected to an existing sea-water heat exchanger to mitigate the tube-side fouling. This technology has been used in many sea-water exchangers worldwide and shows encouraging results.

In addition to the above fouling mitigation technologies, several test and new methods are introduced. Webb and Chamra [67] presented the results of accelerated particulate fouling tests performed on different enhanced tubes and a plain tubes. They concluded that the on-line cleaning systems were very effective in removing fouling. Jardin and Krueger [68] presented the results of using the technology of tube inserts in refinery. They highlighted that these inserts reduced fouling and thus resulted in energy savings, extended run length of exchangers, reduced maintenance cost and enhanced production.

Study Objectives

The overall objectives of this study are summarized as follow:

1. To present and introduce different methods to evaluate fouling in heat exchangers. For exchanger case 1, two years data was used, and for exchanger case 2, one year data was used to evaluate fouling by the proposed methods.

2. To investigate the technical merits of introducing new fouling mitigation technologies (i.e. helically baffled and self-cleaning technologies) in naphtha and crude oil services. The effectiveness of these technologies in mitigating fouling will be evaluated. On-line monitoring of the two mitigation techniques (i.e. new bundles with helical baffles for exchanger case 1 and self cleaning heat exchanger test unit for exchanger case 2) will provide the necessary data for making such an evaluation.

CHAPTER 3

PROBLEM DESCRIPTION

The heat exchangers' fouling is one of the major unresolved problems that affect the exchangers' performance and consequently plants' operation. In this study, different methods to evaluate fouling will be presented and introduced. Also, two fouling mitigation technologies will be presented and their effectiveness will be evaluated. As examples, the analyses and the mitigation technologies will be implemented and tested in two actual exchangers in oil industry. The examples under consideration are naphtha heat exchanger, which has frequent shell side fouling, and thermosyphon reboiler, which has severe tube side fouling. The locations of these exchangers in their associated plants are given in the following sections:

Naphtha Exchanger (Exchanger Case 1)

The naphtha exchanger (exchanger case 1) is located in the recovery plus unit in one of Saudi Aramco refineries. The construction, process and thermal data for exchanger case 1 is shown in Appendix A (Table A1). The recovery plus system as shown in Figure 3.1 is designed to recover reformate (light naphtha) from the net gas (collected gases) by contact with chilled liquid. The net gas stream is pre-cooled with chilled lean gas (from cold separator) in the vapor exchanger. Next, it will be mixed with the lean oil and then sent to the

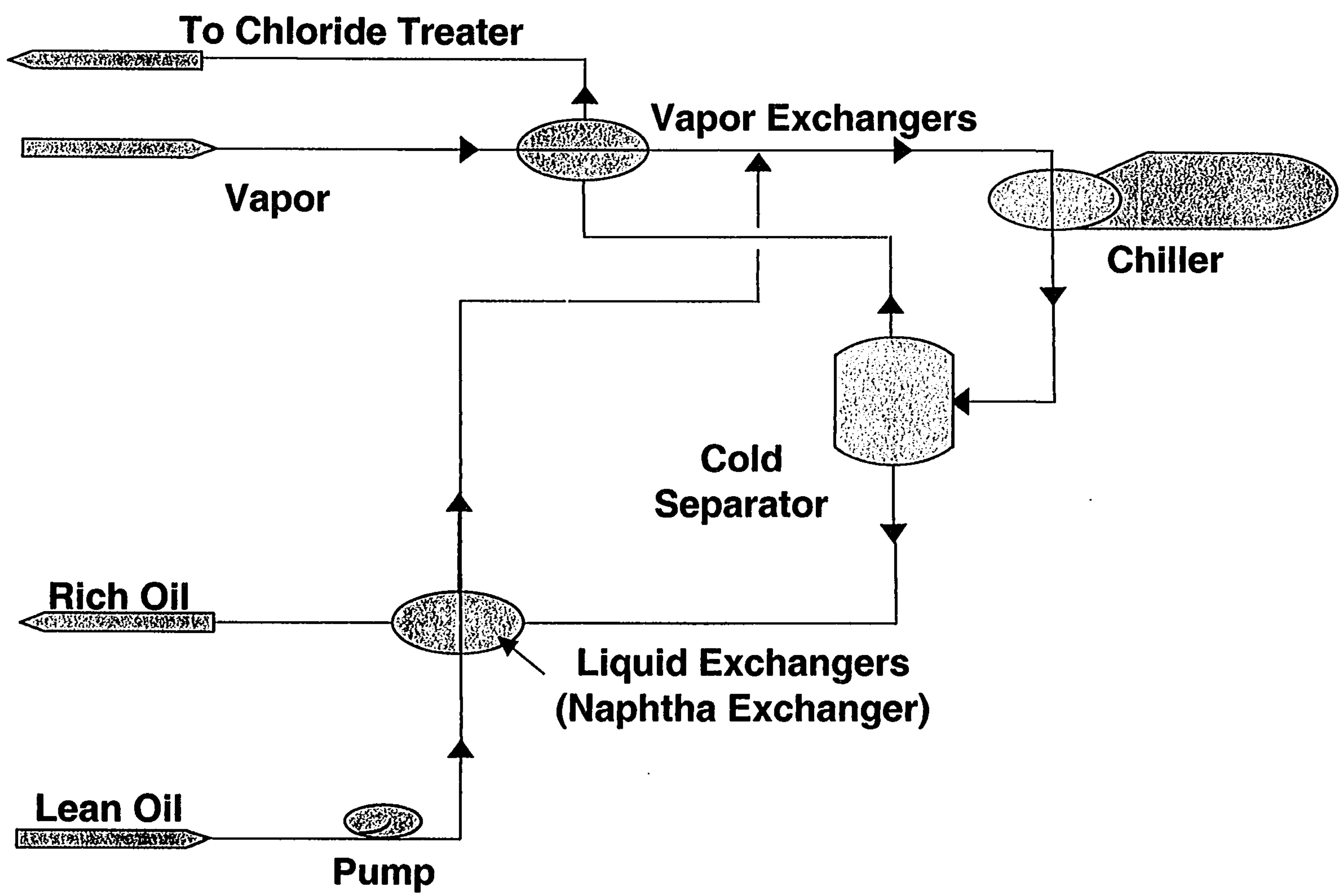


Figure 3.1 Recovery Plus System Which Includes The Naphtha Exchanger (Exchanger Case 1)

chiller. Then, it will be cooled in the chiller and then sent to the cold separator. In the cold separator, the gas goes to the vapor exchanger and then to the chloride treater while the liquid (rich oil) will go to the liquid exchanger (naphtha exchanger). This recovery plus unit recover about 1000 barrel per day. The recovery process upgrades the quality of the recovered product and hence it increases its price by about $ 20 per barrel.

The recovery plus unit experiences a severe fouling in the shell side of the liquid exchanger (naphtha exchanger) as shown in Figure 3.2. The analysis of the deposit (weight percent) is shown in Table 3.1.

Table 3.1: Lab Analysis of the Deposit

Component	Weight Percent
FeS	17%
NaCl	4%
MgCl	1%
$CaSO_4$	40%
$CaCO_3$	2%
$MgCO_3$	2%
$FeCO_3$	4%%
SiO_2	2%
Fe_2O_3	22%

The naphtha heat exchangers (three exchangers in series) are cleaned every two to three months and retubed annually because of severe fouling. Shutting down the recovery plus unit costs about $ 20,000 per day (the cost of the recovered reformate).

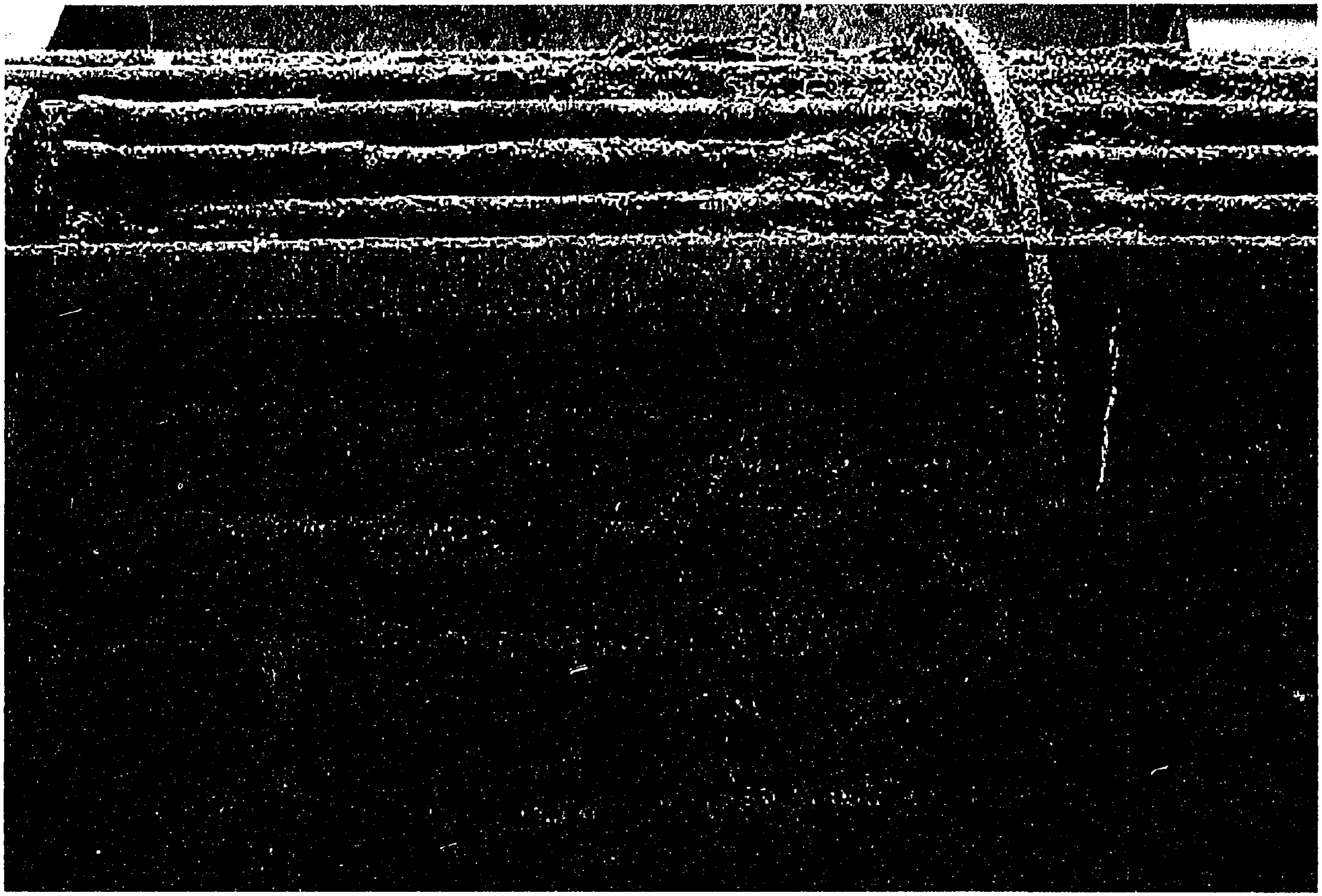


Figure 3.2: Fouling in the Naphtha Heat Exchanger (Exchanger Case 1)

Thermosyphon Reboiler (Exchanger Case 2)

The thermosyphon reboiler is one of many reboilers in one of Saudi Aramco stabilization plants. The construction, process and thermal data for exchanger case 2 is shown in Appendix B (Table B1). The stabilization plant is used to remove hydrogen sulfide (H_2S) from the crude. Crude oil flows into the stabilization plant from different gas oil separation plants (GOSPs) as feedstock crude. In the stabilization plant (Figure 3.3), sour crude with 350-600 parts per million (ppm) of hydrogen sulfide is fed at a controlled rate to the stabilizer columns and flows downward in the column over a series of trays. Oil from the bottom tray enters the reboilers (heat exchangers) where it is heated by steam. The crude oil temperature at inlet is about 70^0 C and outlet temperature is about 85^0 C, and the steam inlet temperature is about 157^0 C and outlet is about 87^0 C. The heated oil returns to the compartment in the base of the column, from which bottom pumps deliver the sweetened crude through the air-cooled heat exchangers to storage tanks or to pump stations and pipeline systems.

Due to the tube side fouling as shown in Figure 3.4, the reboilers (heat exchangers) can not eventually perform the required heat duty owing to the additional resistance to heat transfer. As a result, frequent cleaning is performed to meet the required performance level. They clean the stabilizer columns' heat exchangers every one to two months because of the severe fouling. After almost two years, the deposits become hard deposit, which cannot be cleaned by the normal hydrojeting and consequently, they retube the

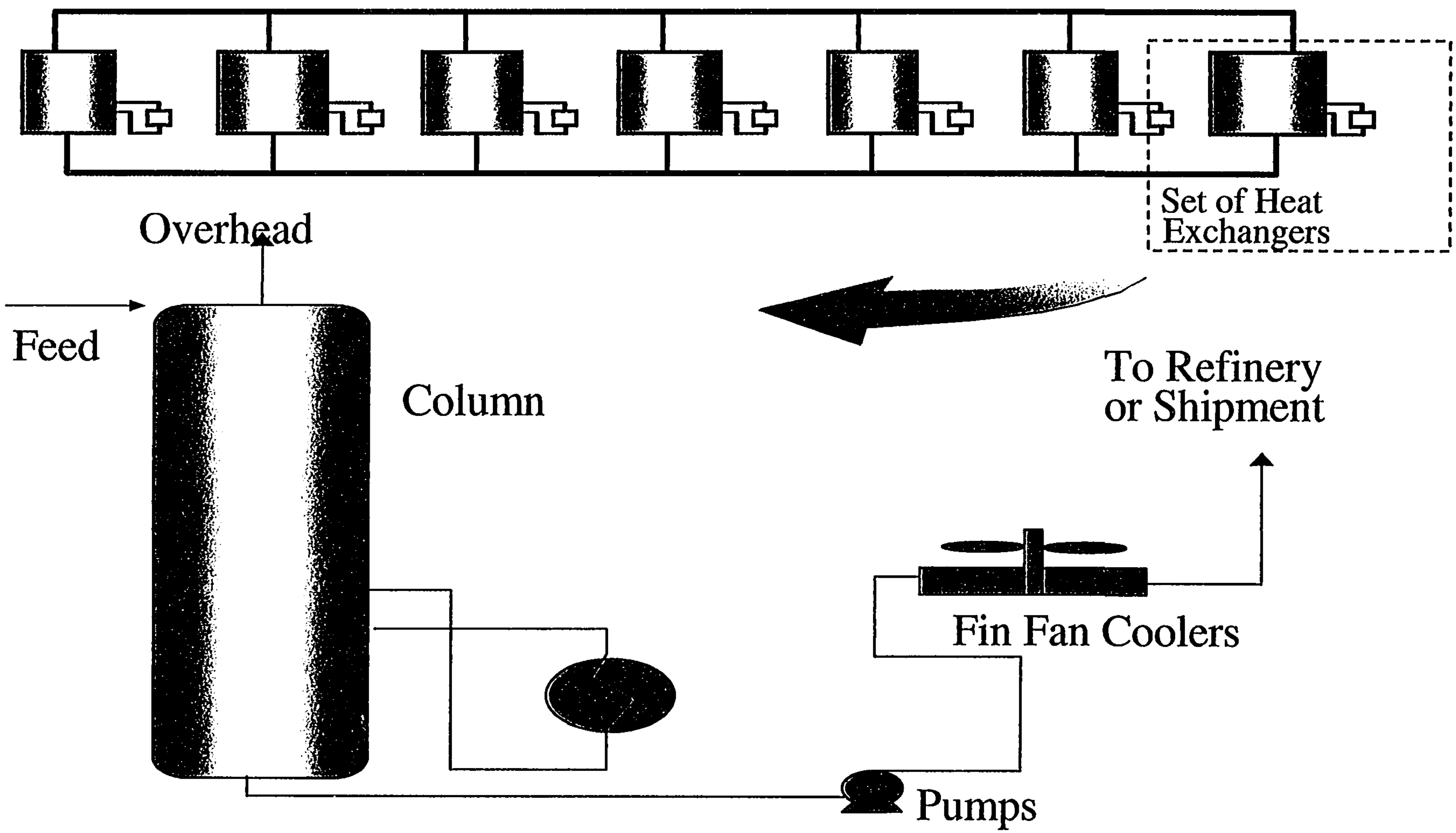


Figure 3.3: Stabilization Plant Which Includes The Thermosyphon Reboiler (Exchanger Case 2)

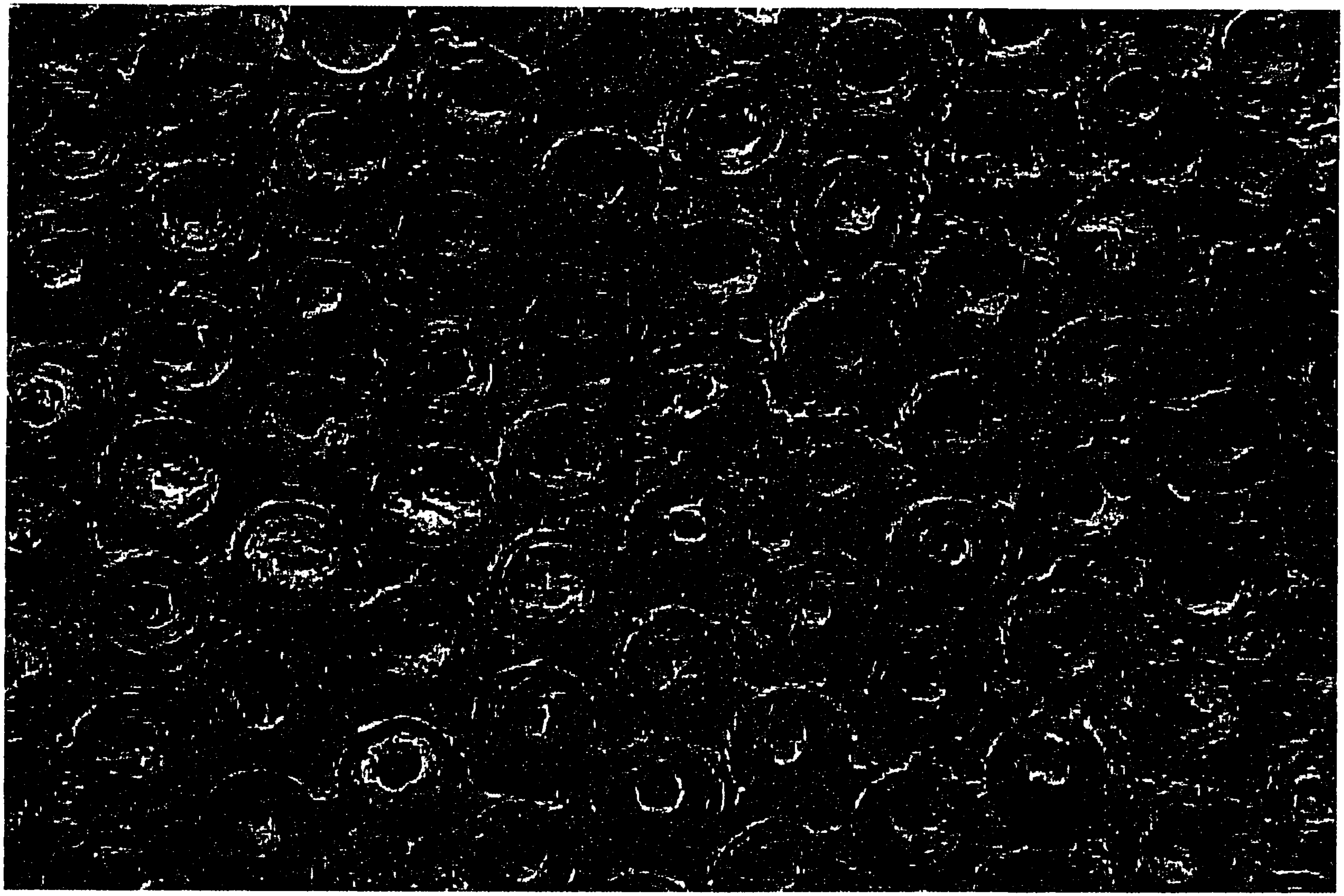


Figure 3.4: Fouling in the Thermosyphon Reboiler (Exchanger Case 2)

heat exchangers frequently. This results in considerable energy losses. Moreover, the total maintenance cost for all reboilers in the stabilization plant (more than 40 reboilers) is about 0.25×10^6 \$/ year.

CHAPTER 4

FOULING CALCULATIONS

The resistance to heat transfer imposed by foulant layer has to be taken into account in the design calculations in order for the heat exchanger to perform its required duty. In general, fouling on a heat transfer surface is a gradual process and the fouling layer develops over a period of time. In this chapter, thermal calculation procedure of fouling is introduced. Then, the online experiments carried out for exchanger case 1 and exchanger case 2 are presented. Next, the solution procedure and the error estimation in calculating fouling are introduced. Finally, the results and discussion for both exchanger case 1 and exchanger case 2 are presented and discussed.

4.1 Thermal Calculation of Fouling

Using heat exchanger thermal evaluation methods, the variation of fouling resistance with time is given as:

$$R_f(t)=\frac{1}{U_a(t)}-\frac{1}{U_c(t)} \tag{4.1}$$

where:

$U_a(t)$: is the actual overall heat transfer coefficient for the heat exchanger at time t.

$U_c(t)$: is the theoretical heat transfer coefficient for the exchanger when it is in clean condition at time t and at the same thermal and hydraulic conditions as U_a. U_c is normally is a constant quantity provided that the flow rates and temperatures are constant with respect to time. However, since the flow rates and temperatures are varying in the present study as a result of process demand, therefore, U_c will be calculated as a function of time.

To determine the values of U_a and U_c, data on fluid characteristics and heat exchanger geometry are required. Such data include: the heat exchanger geometry, inlet and outlet fluid temperatures for both fluid streams, and flow rates and properties.

4.1.1 Calculation of Actual Overall Heat Transfer Coefficient

The actual heat transfer coefficient U_a can be calculated using the energy balance equation [5]:

$$U_a = \frac{\dot{Q}}{A \Delta T_{lm} F} \tag{4.2}$$

The heat transfer rate $\dot{Q}$ can be determined using energy balance on the tube or shell side fluid streams:

$$\dot{Q} = \dot{m}_t c_{p,t} \Delta T_t \tag{4.3}$$

$$\dot{Q} = \dot{m}_s c_{p,s} \Delta T_s \tag{4.4}$$

where:

A is the heat transfer area, m^2

ΔT_{lm} the log mean temperature difference, given by:

$$\Delta T_{\ell m} = \frac{\left(T_{h,in} - T_{c,out}\right) - \left(T_{h,out} - T_{c,in}\right)}{\ln\left[\frac{\left(T_{h,in} - T_{c,out}\right)}{\left(T_{h,out} - T_{c,in}\right)}\right]} \quad \text{for single phase flow} \tag{4.5}$$

$\dot{m}_t$: the tube side flow, kg/s.

$\dot{m}_s$: the shell side flow rate, kg/s.

$c_{p,t}$: the specific heat for tube side, J/kg K.

$c_{p,s}$: the specific heat for the shell side, J/kg K.

ΔT_t : temperature difference for the tube side , K.

ΔT_s : temperature difference for the shell side, K.

F : is the correction factor. It is a function of heat exchanger configuration, thermal effectiveness P and fluid capacitance ratio R.

The thermal effectiveness, P, is given as:

$$P = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} \tag{4.6}$$

and the ratio of fluid capacitance ratio is given by:

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}} \tag{4.7}$$

The correction factor F for 1 shell 2-tube passes (1-2 arrangement) shell and tube heat exchanger is given by [25]:

$$F = \frac{\sqrt{R^2+1}}{1-R} * \frac{\ln\left[\frac{1-PR}{1-P}\right]}{\ln\left[\frac{2-P\left(R+1-\sqrt{R^2+1}\right)}{2-P\left(R+1+\sqrt{R^2+1}\right)}\right]} \tag{4.8}$$

For N shells in series with 2-tube passes in each shell, equation (4.8) can be used to determine F by replacing P with P* which is not equal to P. The modified thermal effectiveness P* is given as [69]:

$$P^* = \frac{\left[\frac{1-PR}{1-P}\right]^{1/N}}{\left[\frac{1-PR}{1-P}\right]^{1/N} - R} \qquad for \qquad R \neq 1 \tag{4.9}$$

$$P^* = \frac{P}{P-NP+N} \qquad for \quad R=1 \tag{4.10}$$

In the case of phase change, the log mean temperature difference should be replaced by the weighted log mean temperature difference $(\Delta T_{\ell n})_{wtd}$, which is given as [69]:

$$(\Delta T_{\ell n})_{wtd} = \frac{\dot{Q}_{tot}}{\sum_i \frac{\dot{Q}_i}{(\Delta T_{\ell n})_i}} \tag{4.11}$$

where:

$\dot{Q}_{tot}$: is the total heat transfer rate, W.

$\dot{Q}_i$: is the heat transfer for phase i, W.

$(\Delta T_{ln})_i$: is the log mean temperature difference for phase i, K.

4.1.2 Calculation of Theoretical (Clean) Overall Heat Transfer Coefficient

The theoretical (clean) heat transfer coefficient U_c is calculated using the following expression [5]:

$$U_c = \frac{1}{\frac{1}{h_s} + \frac{D_o}{2k}\ln\left(\frac{D_o}{D_i}\right) + \frac{D_o}{h_t D_i}} \tag{4.12}$$

The tube side heat transfer coefficient can be determined by using a number of well-established correlations for internal flows in conduits reported in the literature [5, 25, 70]. One such correlation for turbulent flow in smooth pipes is given as:

$$Nu_t = 0.023 \ \mathrm{Re}_t^{0.8} \ \mathrm{Pr}_t^{0.4} \tag{4.13}$$

where:

$$Nu_t = \frac{h_t D_i}{k_t} \tag{4.14}$$

$$\mathrm{Re}_t = \frac{\rho_t V_t D_i}{\mu_t} \tag{4.15}$$

$$\Pr_t = \frac{\mu_t C_{p,t}}{k_t} \qquad 4.16)$$

The correlation of the shell side heat transfer coefficient for the case of condensation in the shell side, h_{cond}, were developed in the literature [25]. It is given by:

$$h_{cond} = 0.943\left(\frac{\rho_l g(\rho_l - \rho_v)k_l^3 h_{lv}}{\mu_l (T_{sat} - T_{sur})L_{cond}}\right)^{0.25} \qquad (4.17)$$

where:

k_l	: Thermal conductivity for the liquid, W/ m K
μ_l	: Viscosity for the liquid, N s/ m^2
ρ_l	: Density for the liquid, kg/m^3
ρ_v	: Density for the vapor, kg/m^3
h_{lv}	: Latent heat of evaporation, J/kg
g	: Gravitational acceleration, 9.81 m^2/s
T_{sat}	: Saturated temperature, °C
T_{sur}	: Surface temperature, °C
L_{cond}	: Tube length for condensation, m

Because of the complex flow configurations, calculations of the shell side heat transfer coefficient and pressure loss are difficult. The calculation procedures have evolved over the last 50 years. The well-established methods include Kern [5], Bell Delaware (BD) [5, 71] and Flow Stream Analysis (FSA)

[5, 72] methods. Since the Bell Delaware and Flow Stream Analysis methods will be used in this study, the detailed calculation procedure for each method is outlined below for the sake of completeness and so that the write up can stand by itself. Also, since the calculations of the shell side heat transfer coefficient is more involved as compare to the calculations of the tube side heat transfer coefficient, the thermal analysis methods used in this study will be called either BD method or FSA method.

Bell Delaware Method

The Bell Delaware method developed in 1963 introduced factors for baffle leakage, and all effects on the shell side. The correction factors were based on extensive experimental data. The Bell Delaware method is extensively used. Considering a shell and tube heat exchanger with the flowing geometry:

Shell internal diameter : D_s, m

Number of tubes : N_T

Tube outside diameter : D_o, m

Tube inside diameter : D_i, m

Tube pitch : P_T, m

Baffle spacing : L_B, m

Shell length : L_s, m

Tube to baffle diametral clearance : Δ_{tb}, m

Shell to baffle diametral clearance : Δ_{sb}, m

Bundle to shell diametral clearance : Δ_b, m

Number of sealing strips per cross flow raw : N_{ss}/N_c

Baffle cut : B_c

Thickness of baffles : t_b, m

Number of tube side passes : n

The first step to calculate the shell side heat transfer coefficient that is given by:

$$h_s = h_c J_c J_L J_B \tag{4.18}$$

Where h_c the ideal cross flow heat transfer coefficient is given by the following correlation [5]:

$$h_c = \frac{k_s}{D_o}\left(0.211 \mathrm{Re}_s^{\ 0.651} \mathrm{Pr}_s^{\ 0.34}\right) \tag{4.19}$$

The Reynolds number and Prandtl number are given by:

$$\mathrm{Re}_s = \frac{\rho_s V_{max} D_o}{\mu_s} \tag{4.20}$$

$$\mathrm{Pr}_s = \frac{\mu_s C_p}{k_s} \tag{4.21}$$

And the maximum intertube velocity: V_{max} is given as:

$$V_{max} = \frac{\dot{m}_s}{\rho_s S_m} \tag{4.22}$$

where:

$\dot{m}_s$ is the total mass flow rate in the shell, kg/ s

ρ_s is the shell side fluid density, kg/m^3

$C_{p,s}$ is the shell side specific heat, J/kg K

μ_s is the shell side viscosity, N s/m^2

S_m is the flow area near the centerline and is given as:

$$S_m = L_B \left[D_s - D_{OTL} + \frac{D_{OTL} - D_o}{(aa)\ P_T} (P_T - D_o) \right] \quad (4.23)$$

aa = 1 for square (90°) pitch [5];

aa = 0.866 for triangular (30) pitch

aa = 0.707 for rotated squares (45°) pitch

aa = 0.5 for rotated triangular (60°) pitch

D_{OTL} is the bundle diameter (in m), given by:

$$D_{OTL} = D_s - \Delta_b \quad (4.24)$$

N number of baffles in the shell is given as:

$$\left(N = \frac{L_s}{L_B + t_b} - 1 \right) : \text{ rounded to the nearest integer} \quad (4.25)$$

The correction factor for baffle configuration J_c is given as [5]:

$$J_c = 0.55 + 0.72 F_c \quad (4.26)$$

Where F_s is the fraction of tubes in cross flow, which is given by [5]:

$$F_c = \frac{1}{\pi} \left[\pi + \frac{2(D_s - 2L_c)}{D_{OTL}} \sin\left(\cos^{-1}\left(\frac{D_s - 2L_c}{D_{OTL}} \right) \right) - 2COS^{-1}\left(\frac{D_s - 2L_c}{D_{OTL}} \right) \right] \quad (4.27)$$

L_c is the baffle cut distance and is given by:

$$L_c = \frac{B_c D_s}{100} \quad (4.28)$$

The leakage correction factor J_L is given by the following equation, which is developed in this study using regression methods using the graphical data presented by Hewitt et. al [5]:

$$J_L = 1.0277 - 1.107 X_1 + 0.0049 Y_1 + 0.625 X_1^2 - 0.484 X_1 Y_1 - 0.0137 Y_1^2 \tag{4.29}$$

where X_1 and Y_1 are defined by:

$$X_1 = \frac{S_{sb} + S_{tb}}{S_m} \tag{4.30}$$

$$Y_1 = \frac{S_{sb}}{S_{sb} + S_{tb}} \tag{4.31}$$

where the leakage area (shell-to-baffle) S_{sb} and the leakage area (tube to baffle) S_{tb} are given by [5]:

$$S_{sb} = \frac{D_s \Delta_{sb}}{2}\left[\pi - \cos^{-1}\left(1 - \frac{2L_c}{D_s}\right)\right] \tag{4.32}$$

$$S_{tb} = \frac{\pi D_o \Delta_{tb}}{2} N_T \frac{(1 + F_c)}{2} \tag{4.33}$$

The bypass correction factor (J_B), which is developed in this study using regression by utilizing the graphical data presented by Hewitt et, al. [5], is given by:

$$J_B = 0.9579 - 0.7725 X_2 + 0.6113 Y_2 + 0.136 X_2^2 + 1.4814 X_2 Y_2 - 1.1418 Y_2^2 \tag{4.34}$$

where X_2 and Y_2 are defined by:

$$X_2 = \frac{S_b}{S_m} = F_{bp} \tag{4.35}$$

$$Y_2 = \frac{N_{ss}}{N_c} \tag{4.36}$$

The bypass area fraction F_{bp} can be expressed by [5]:

$$F_{bp} = \frac{(D_s - D_{OTL})}{S_m} L_B \tag{4.37}$$

The second step is the calculations of the cross flow pressure drop on the shell side, is given by [5]:

$$\Delta P_s = [(N-1)\Delta P_c\, R_B + N\,\Delta P_w]\, R_L + 2\Delta P_c R_B\left(1 + \frac{N_{cw}}{N_c}\right) \tag{4.38}$$

where R_L and R_B are the leakage and bypass correction factors for pressure drop. The equations for these factors which are developed in this study using regression and the data presented by Hewitt et. al [5], are given as:

$$\begin{aligned} R_L &= 1.0404 - 2.3116X_1 - 0.2098Y_1 + 1.8607X_1^2 \\ &+ 0.18\,X_1Y_1 + 0.0606Y_1^2 \end{aligned} \tag{4.39}$$

$$\begin{aligned} R_B &= 0.9242 - 1.7241X_2 + 0.7168Y_2 + 0.5755X_2^2 \\ &+ 0.26615\,X_2Y_2 - 1.0465Y_2^2 \end{aligned} \tag{4.40}$$

To calculate the other factors in equation (4.38), Euler number K_f should be calculated first. It is given for Square arrays (90°) by the following equation [5]:

$$K_f = 0.272 + \frac{0.207\times10^3}{\mathrm{Re}_s} + \frac{0.102\times10^3}{\mathrm{Re}_s^{\ 2}} - \frac{0.286\times10^3}{\mathrm{Re}_s^{\ 3}} \tag{4.41}$$
$$for\ 3 < \mathrm{Re}_s < 2\times10^3$$

or

$$K_f = 0.267 + \frac{0.249\times10^4}{\mathrm{Re}_s} - \frac{0.927\times10^7}{\mathrm{Re}_s^{\ 2}} + \frac{0.10\times10^{11}}{\mathrm{Re}_s^{\ 3}}$$
$$for\ \ 2\times10^3 < \mathrm{Re}_s < 2\times10^6 \tag{4.42}$$

For triangular arrays (30°), it is given by:

$$K_f = 0.795 + \frac{0.247\times10^3}{\mathrm{Re}_s} + \frac{0.335\times10^4}{\mathrm{Re}_s^{\ 2}}$$
$$-\frac{0.155\times10^4}{\mathrm{Re}_s^{\ 3}} + \frac{0.241\times10^4}{\mathrm{Re}_s^{\ 4}}$$
$$for\ 3 < \mathrm{Re}_s < 10^3 \tag{4.43}$$

or

$$K_f = 0.245 + \frac{0.339\times10^4}{\mathrm{Re}_s} - \frac{0.984\times10^7}{\mathrm{Re}_s^{\ 2}}$$
$$+\frac{0.133\times10^{11}}{\mathrm{Re}_s^{\ 3}} - \frac{0.599\times10^{13}}{\mathrm{Re}_s^{\ 4}} \tag{4.44}$$
$$for\ \ 10^3 < \mathrm{Re}_s < 10^6$$

The cross flow pressure drop is given by [5]:

$$\Delta P_c = N_c \mathrm{K}_f \left(\tfrac{1}{2}\, \rho_s V_{\max}^2 \right) \tag{4.45}$$

where the number of cross flow rows N_c is given as:

$$N_c = \frac{D_s(1-2L_c/D_s)}{P_{T_p}} \tag{4.46}$$

And

P_{T_p} : P_T for square tube arrays (90°)

P_T : 0.866 P_T for triangular (30°).

P_T : 0.707 P_T for rotated square (45°)

P_T : 0.5 P_T for rotated triangular (60°)

The window zone pressure drop is given by [5]:

$$\Delta p_w = \frac{(2+0.6N_{cw})\dot{m}_s^2}{2S_m S_w \rho} \tag{4.47}$$

Where the window zone flow area S_w is given by [5]:

$$S_w = \frac{D_s^2}{4}\left[\cos^{-1}\left(\frac{D_s-2L_c}{D_s}\right) - \left(\frac{D_s-2L_C}{D_S}\right)\sqrt{\left[1-\left(\frac{D_s-2L_c}{D_s}\right)^2\right]}\right] - \frac{N_T}{8}(1-F_c)\,\pi D_o^2 \tag{4.48}$$

and the effective number of cross flow rows in window zone, N_{cw}, is given by:

$$N_{cw} = \frac{0.8L_c}{P_{Tp}} \tag{4.49}$$

<u>Flow Stream Analysis Method</u>

The difficulty in using the correction factors, as developed in Bell-Delaware method, for a wide range of configuration lead to the development of the Flow

Stream Analysis method [5, 72]. This technique is particularly suitable for computer calculations. Hence, it has been the basis of many commercial computer codes [5]. For the flow stream analysis method, the input data will be same as those in the BD method. The calculation procedures are as follows:

The shell side heat transfer coefficient h_s is given by:

$$h_s = \frac{k_s}{D_o}\left[0.211\ \mathrm{Re}_s^{0.651}\ \mathrm{Pr}_s^{0.34}\right] \tag{4.50}$$

And the shell side pressure drop, ΔP_s, is given as:

$$\Delta P_s = (N+1)\,\Delta P \tag{4.51}$$

where:

ΔP is the pressure drop for one baffle and is given by:

$$\Delta P = n_p \dot{m}_s^2 \tag{4.52}$$

N is the number of baffles in the shell and is given by equation (4.25)

Reynolds number Re_s and Prandtl number Pr_s are calculated as follow:

$$\mathrm{Re}_s = \frac{D_o \dot{m}_s F_{cr}}{\mu_s S_m} \tag{4.53}$$

$$\mathrm{Pr}_s = \frac{\mu_s\ C_{p,s}}{k_s} \tag{4.54}$$

The faction of cross flow F_{cr} and n_p are calculated by iteration as follows:

Assume F_{cr}=0.5

Then, calculate Euler Number K_f by the following equation:

For square arrays (90°)

$$K_f = 0.272 + \frac{0.207\times10^3}{\mathrm{Re}_s} + \frac{0.102\times10^3}{\mathrm{Re}_s^{\ 2}} - \frac{0.286\times10^3}{\mathrm{Re}_s^{\ 3}} \tag{4.55}$$

$$for \qquad 3<\mathrm{Re}_s<2\times10^3$$

or

$$K_f = 0.267 + \frac{0.249\times10^3}{\mathrm{Re}_s} - \frac{0.927\times10^7}{\mathrm{Re}_s^{\ 2}} + \frac{0.1\times10^{11}}{\mathrm{Re}_s^{\ 3}} \tag{4.56}$$

$$for \quad 2\times10^3 < \mathrm{Re}_s < 2\times10^6$$

For triangular arrays (30°)

$$K_f = 0.795 + \frac{0.247\times10^3}{\mathrm{Re}_s} + \frac{0.335\times10^4}{\mathrm{Re}_s^{\ 2}} - \frac{0.155\times10^4}{\mathrm{Re}_s^{\ 3}} + \frac{0.241\times10^4}{\mathrm{Re}_s^{\ 4}} \tag{4.57}$$

$$for \quad 3<Re_s<10^3$$

or

$$K_f = 0.245 + \frac{0.339\times10^4}{\mathrm{Re}_s} - \frac{0.984\times10^7}{\mathrm{Re}_s^{\ 2}} + \frac{0.133\times10^{11}}{\mathrm{Re}_s^{\ 3}} - \frac{0.599\times10^{11}}{\mathrm{Re}_s^{\ 4}} \tag{4.58}$$

$$for \quad 10^3<\mathrm{Re}_s<10^6$$

Next, calculate n_c as shown below [5]:

$$n_c = \frac{N_c K_f}{2\rho S_m^2} \tag{4.59}$$

Then, calculate n_{cb} as shown below [5]:

$$n_{cb} = \left(n_c^{-1/2} + n_b^{-1/2}\right)^{-2} \tag{4.60}$$

Next, calculate n_a as shown below [5]:

$$n_a = n_w + n_{cb} \tag{4.61}$$

Then, calculate n_p as shown below [5]:

$$n_p = \left(n_a^{-1/2} + n_s^{-1/2} + n_t^{-1/2}\right)^{-2} \tag{4.62}$$

Finally, calculate the new value of F_{cr} as given by [5]:

$$F_{cr} = \left(\frac{n_p}{n_a}\right)^{1/2}\left[1 + \left(\frac{n_c}{n_p}\right)^{1/2}\right]^{-1} \tag{4.63}$$

where:

S_s is the shell-to-baffle leakage area and given by [5]:

$$S_s = \pi\left(D_s - \frac{\Delta_{sb}}{2}\right)\frac{\Delta_{sb}}{2} \tag{4.64}$$

n_s is the shell-to-baffle leakage resistance and given by [5]:

$$n_s = \frac{0.036\left(\frac{2t_b}{\Delta_{sb}}\right) + 2.3\left(\frac{2t_b}{\Delta_{sb}}\right)^{-0.177}}{2\rho S_s^2} \tag{4.65}$$

S_t is the tube to baffle leakage area and is given by [5]:

$$S_t = N_T\pi\left(D_o + \frac{\Delta_{tb}}{2}\right)\frac{\Delta_{tb}}{2} \tag{4.66}$$

n_t is the tube to baffle leakage resistance and is given by:

$$n_t = \frac{0.036\left(\frac{2t_b}{\Delta_{tb}}\right) + 2.3\left(\frac{2t_b}{\Delta_{tb}}\right)^{-0.177}}{2\rho S_t^2} \tag{4.67}$$

D_{OTL} is the tube bundle diameter (in m) and is given by:

$$D_{OTL} = D_S - \Delta_b \tag{4.68}$$

S_m is the flow area near the center line which can be expressed by [5]:

$$S_m = L_B\left[D_s - D_{OTL} + \frac{D_{OTL} - D_o}{(aa)\,p_T}(p_T - D_o)\right] \tag{4.69}$$

where:

aa = 1 for square (90°) pitch

aa = 0.866 for triangular (30°) pitch

aa = 0.707 for rotated square (45°) pitch

aa = 0.5 for rotated triangular (60°) pitch

F_c is the fraction of tubes in cross flow and is given by [5]:

$$F_c = \frac{1}{\pi}\left[\pi + \frac{2(D_s - 2L_c)}{D_{OTL}} \sin\left(\cos^{-1}\left(\frac{D_s - 2L_c}{D_{OTL}}\right)\right) - 2\cos^{-1}\left(\frac{D_s - 2L_c}{D_{OTL}}\right)\right] \quad (4.70)$$

where: L_c is the baffle cut distance, m

$$L_c = \frac{B_c D_s}{100} \quad (4.71)$$

S_w is the window zone flow area and is given by [5]:

$$S_w = \frac{D_s^2}{4}\left[\cos^{-1}\left(\frac{D_s - 2L_c}{D_s}\right) - \left(\frac{D_s - 2L_C}{D_S}\right)\sqrt{\left[1 - \left(\frac{D_s - 2L_c}{D_s}\right)^2\right]}\right] - \frac{N_T}{8}(1 - F_c)\,\pi D_o^2 \quad (4.72)$$

n_w is the window flow resistance and given by [5] as:

$$n_w = \frac{1.9\exp(0.6856\ S_w / S_m)}{2\rho\ S_w^2} \quad (4.73)$$

S_b is the bypass flow area and given by [5] as:

$$S_b = 2\frac{\Delta_b}{2} L_B \quad (4.74)$$

N_c is the number of cross flow rows and given by [5] as:

$$N_c = \frac{D_s(1-2L_c/D_s)}{P_{Tp}} \tag{4.75}$$

where:

$P_{TP} = P_T$ for square arrays (90°)

$P_{TP} = 0.866\ P_T$ for (30°)

$P_{TP} = 0.707\ P_T$ for (45°)

$P_{TP} = 0.5\ P_T$ for (60°)

N_{ss} is the effective number of sealing strips and is calculated by the following equation [5]:

$$N_{ss} = N_c * (N_{ss}/N_c) \tag{4.76}$$

n_b is the bypass flow resistance, expressed as:

$$n_b = \frac{0.266N_c + N_{ss}}{2\rho S_b^2} \tag{4.77}$$

After converging of the above iteration to calculate F_{cr}, the following equations could be used to calculate the bypass flow F_b, the shell-to-baffle leakage flow F_s, and the tube-to-baffle leakage flow F_t :

$$F_b = \left(\frac{n_p}{n_a}\right)^{½}\left[1+\left(\frac{n_b}{n_c}\right)^{½}\right]^{-1} \tag{4.78}$$

$$F_s = \left(\frac{n_p}{n_s}\right)^{½} \tag{4.79}$$

$$F_t = \left(\frac{n_p}{n_t}\right)^{½} \tag{4.80}$$

4.1.3 Solution Procedure

A number of computer programs written to determine fouling resistance in heat exchangers are available. Some of these programs are based on a number of assumptions that simplify the solution and hence it affects the accuracy of the results. Such programs do not require detailed information about the heat exchanger. On the other hand there are computer packages that require detailed information and the results are very accurate for a wide range of geometrical and operational parameters. These packages, which are based on the thermal analysis method for calculating fouling resistances, are proprietary, expensive, and not published in open literature.

In this study, the investigator has written his own computer programs to solve the set of algebraic equations that resulted from the above BD and FSA methods.

The flow charts of the developed computer programs for the BD and FS analysis methods are shown in Figures 4.1 and 4.2, respectively. The input data included: inlet and outlet temperatures, flow rates, properties for both fluids and detailed geometry of the heat exchanger under consideration. Both programs calculate the actual and the clean overall heat transfer coefficients. These coefficients are used to determine the fouling resistance for the subject heat exchanger.

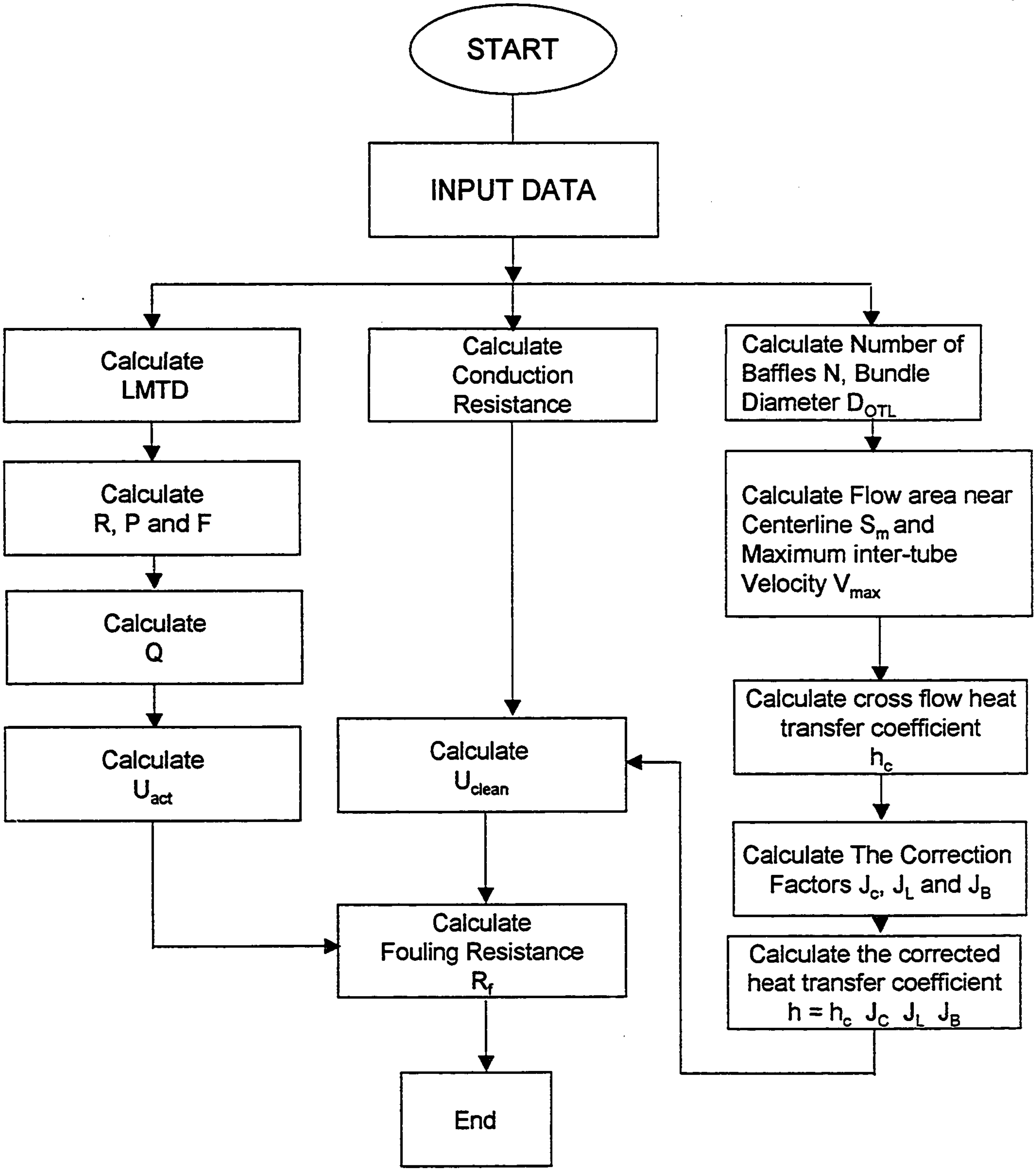


Figure 4.1: Flow Chart for Fouling Resistance Calculation Using Bell-Delaware Method

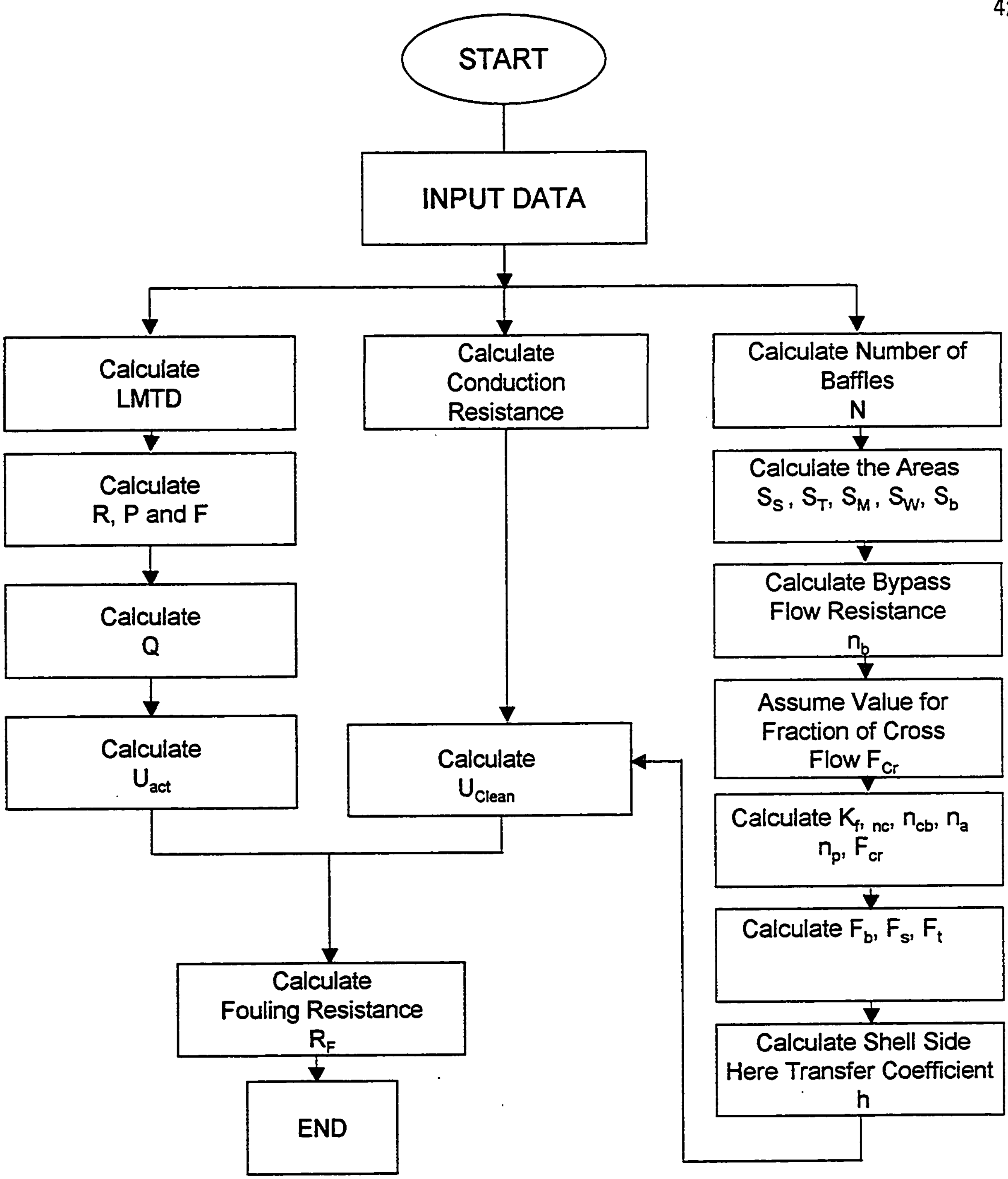


Figure 4.2: Flow Chart for Fouling Resistance Calculations Using Flow Stream Analysis

4.2 On-Line Experiments

On line experiments were conduced to study fouling growth in both the naphtha shell-and-tube heat exchanger (exchanger case 1) in the recovery plus plant and the thermosyphon reboiler (Exchanger case 2) in the stabilization plant as discussed in chapter 3. More than two years data is collected for exchanger case 1 and more than one year data is collected for exchanger case 2. The collected data include inlet and outlet temperatures and mass flow rates for both the shell-and-tube sides. More information is provided for the instruments used in the experiments in the following paragraphs:

Exchanger Case 1:

To measure the inlet and outlet temperatures for the naphtha heat exchanger, temperature transmitters of type RTD (Resistance Temperature Detector)[73] are used. RTD is a general description for any device that senses temperature by a variation in the resistance of an electrically conductive material. RTDs are the most accurate method of measuring temperature over wide ranges and are highly stable over time and temperature cycling [74]. The conductive material is most often platinum. The most common configuration of an RTD is a probe form as shown in Figure 4.3. An RTD probe consists of a protective sheath, which is often a closed-end stainless steel tube, a sensor element, lead wires, and a threaded termination. Although an RTD probe has a protective sheath, it is inserted into a thermowell, as shown in Figure 4.4, for added protection from process contaminants.. The temperature range for both

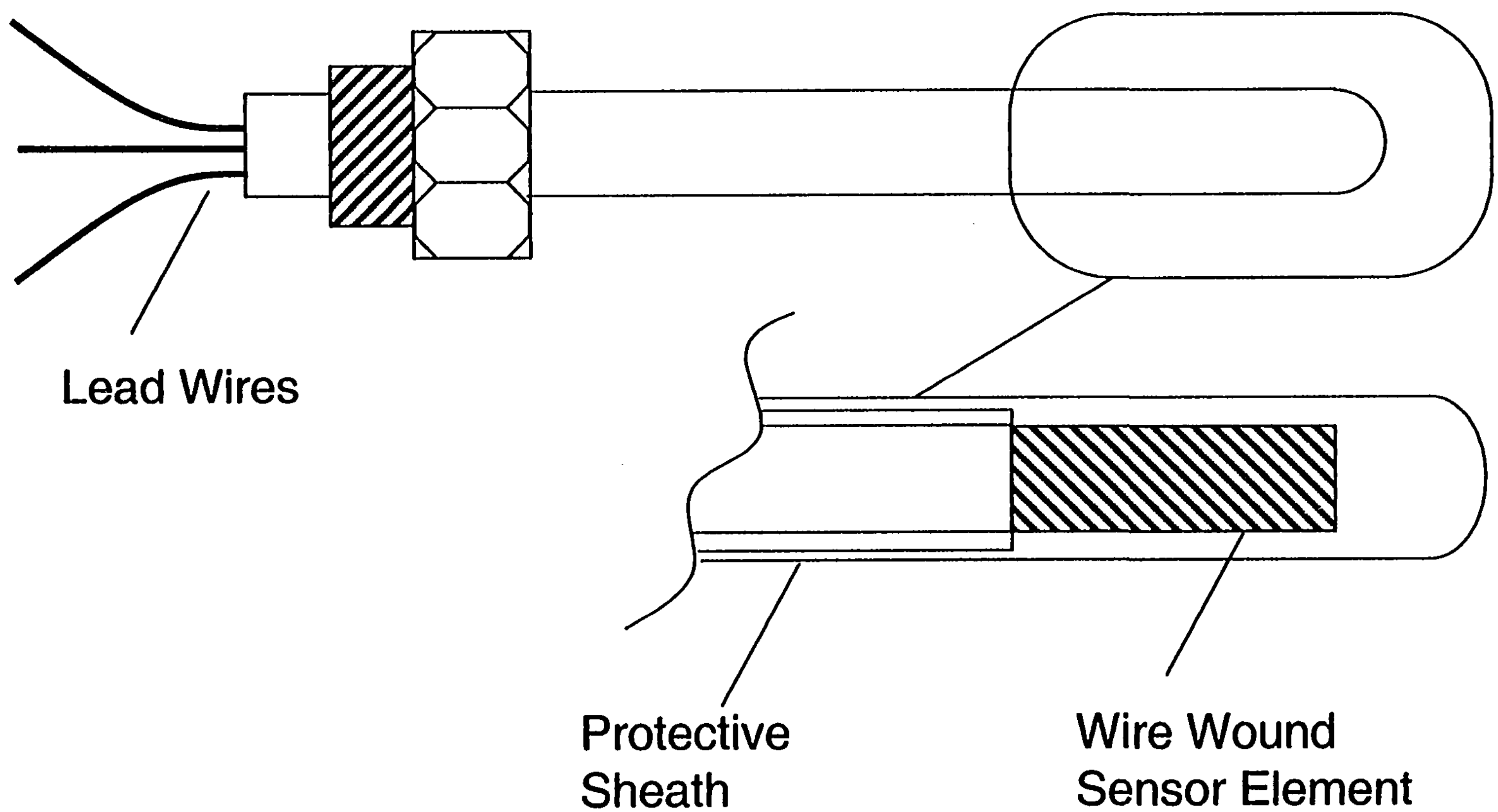


Figure 4.3: RTD Probe Construction

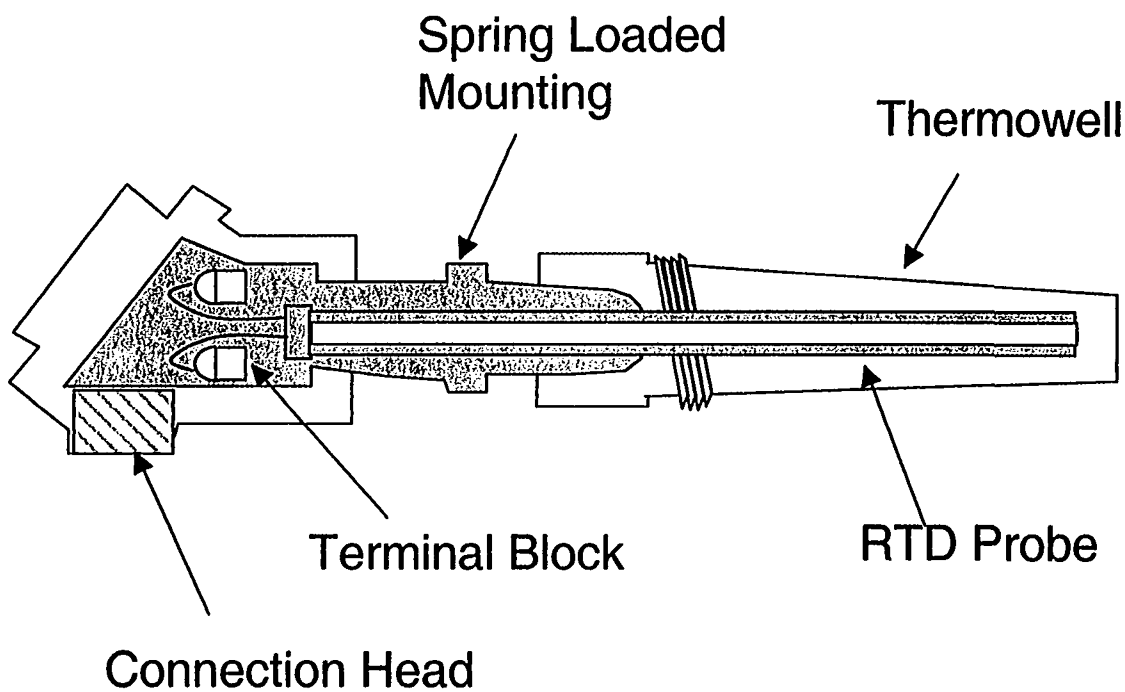


Figure 4.4: RTD Inserted in Thermowell

the shell and tube sides is from -130 ° C to 300° C. The inaccuracy in this temperature range is about 1.5% as provided in the instrument manual. However, based on the instrument engineers' handbook [74], the inaccuracy is about 0.3%. Moreover, this accuracy of the temperature transmitter was checked in the lab using thermometer and the results are shown in Table 4.1. It could be seen from the table that the maximum difference in the reading is about 1%. To be conservative, the maximum error, which the one provided in the instrument manual, will be used in the error estimation.

Table 4.1: Comparison of the Temperature Readings

RTD	Thermometer	% Difference
75.1	75.4	0.40
80.6	81.2	0.74
85.7	85.9	0.23
90.1	91.0	0.99
95.2	95.9	0.73
98.5	98.7	0.20

Flow rates for both the shell and tube sides were measured using flow transmitters of differential type (flow nozzle) [73]. The operation of such flow nozzles follows Bernoulli's theorem. A typical nozzle installation is shown in Figure 4.5. The nozzle design allows for a smooth entry and sharp exit as shown in Figure 4.6. The nozzle can be welded to the pipeline, or mounted in a holding ring between flanges. If frequent maintenance or inspection of flow nozzle is required, then the holding ring design is preferred. Flow nozzles can be fabricated from any material such as aluminum, fiberglass, stainless steel or chrome-moly steel [74]. The manufacturer of the differential type flow nozzle for the naphtha heat exchanger is Foxboro. The measurement range is from 0

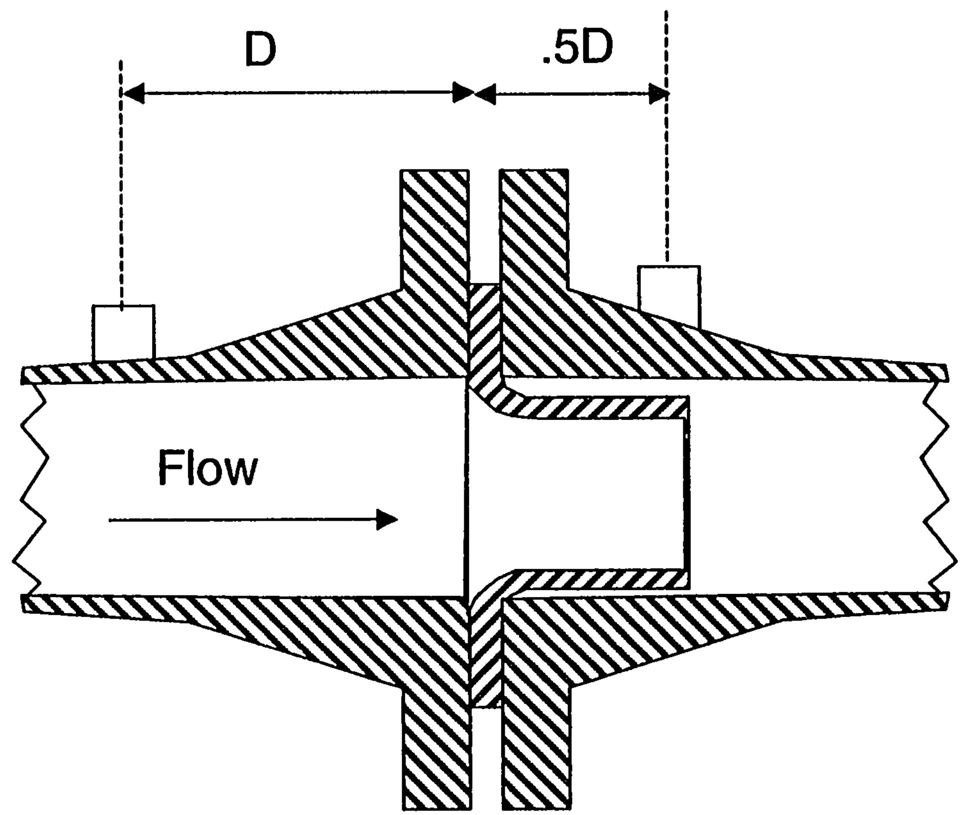
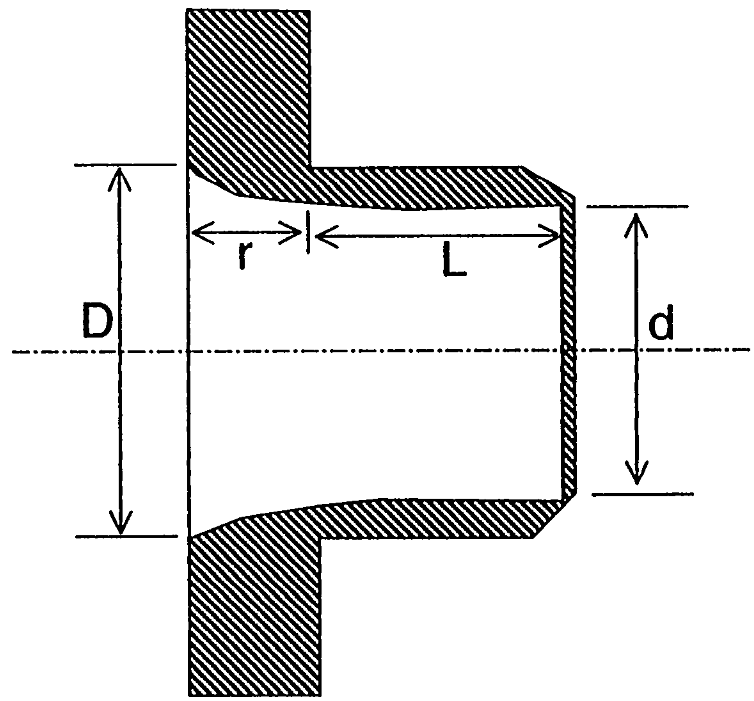


Figure 4.5: Typical Nozzle Installation



$r = 0.5\ D$
$L < 0.6\ d$ or $L < 0.33\ D$

Figure 4.6: Flow Nozzle Constructions

kg/s to 100 kg/s with inaccuracy of about 1.5% as provided in the instrument manual. Moreover, based on the instrument engineers' handbook [74], the inaccuracy is about 1.5%.

The temperature and flow transmitters are connected to the control room for continuous monitoring and data logging. More than 2 years of data were collected. The data included inlet and outlet temperatures and mass flow rates for both fluid streams. Samples of these data are shown in Table 4.2 and Appendix A.

Table 4.2: Samples of the Collected Data for Exchanger Case 1

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	4/19/99	41.83	-5.71	-17.65	24.71	10.47	9.70
3	4/21/99	43.36	-5.85	-17.69	24.27	11.90	10.55
5	4/23/99	41.04	-5.62	-17.72	23.35	11.80	10.80
7	4/25/99	44.00	-5.95	-17.71	22.79	12.81	10.80
9	4/27/99	43.50	-5.98	-17.67	22.66	12.75	10.80

Exchanger Case 2:

The instruments that were used to collect the required data from the reboiler are temperature and flow transmitters. Similar RTD type used in the naphtha heat exchangers are used in the reboiler [75]. However, the temperature ranges are different. For the crude (tube side), the temperature range for the temperature transmitter is from 38° C to 150° C, whereas, the temperature range for the steam/condensate (shell side) is from 38° C to 540° C. The inaccuracy is about 1.5% as per the instrument manual. This value is conservative value as compared to the value reported in instrument engineers' handbook [74]. Therefore, 1.5% will be used in the calculations.

The flow transmitter for the shell side (condensate) is a differential type (flow nozzle) [75]. The flow range is from 0 kg/s to 100 kg/s and the inaccuracy is about 1.0% as reported in the instrument manual. For the crude, the flow transmitter is differential pressure orifice type. The orifice plate is the most economical type of all differential pressure flow meters. An orifice plate is constructed as a thin, concentric, flat metal plate. The plate has an opening (orifice). The orifice plate is installed perpendicular to the fluid flow between two flanges of a pipe. As the fluid passes through the orifice, the restriction causes an increase in fluid velocity and a decrease in pressure. The potential energy (static pressure) is converted into kinetic energy (velocity). As the fluid leaves the orifice, fluid velocity decreases and pressure increases as kinetic energy is converted back into potential energy (static pressure). Orifice plates always experience some energy loss, which is a permanent pressure loss, caused by the friction in the plate. The most common holding system for an orifice plate is a pair of flanges and pressure taps as shown in Figure 4.7. The pressure tapes are located either on orifice flanges or upstream and downstream of the pipe from the orifice plate (this is the case for the reboiler) [74]. The flow range of the orifice plate used for crude side of the reboiler is from 0 kg/s to 1500 kg/s with inaccuracy of about 1.0% as per the instrument manual.

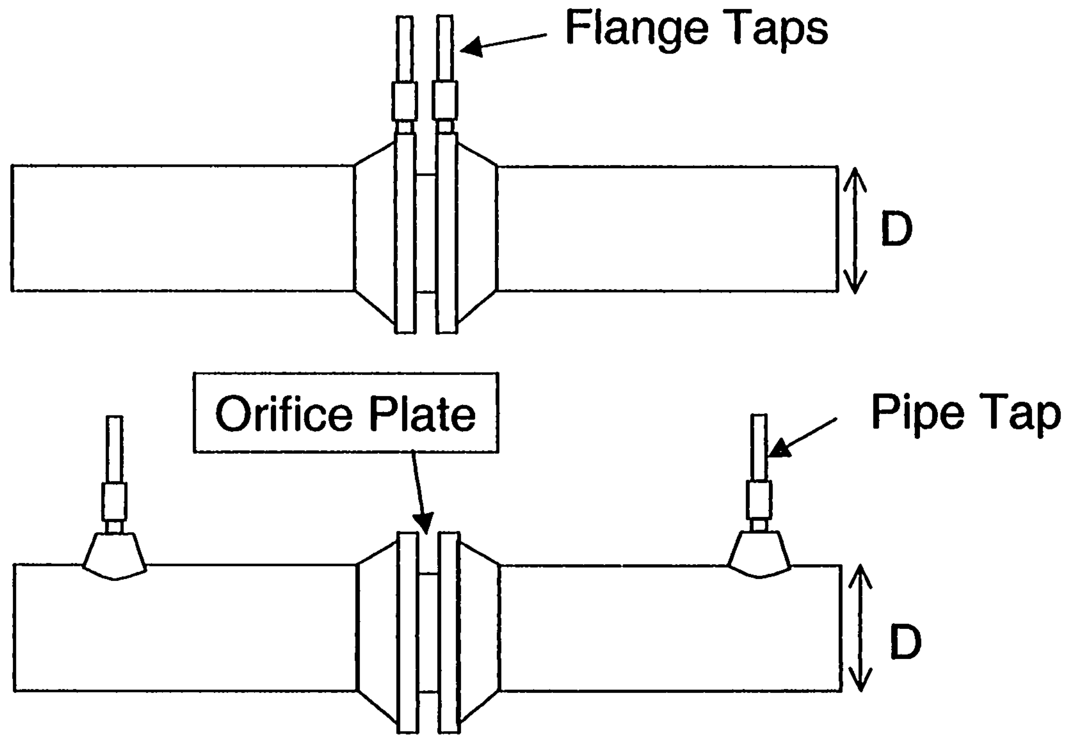


Figure 4.7: Orifice Plate Pressure Tap Configurations

These temperatures and flow transmitters used in the reboiler are connected to the control room for continuous monitoring and data logging. Samples of these data are shown in Table 4.3 and Appendix B.

Table 4.3: Samples of the Collected Data for Exchanger Case 2

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	10/15/99	156.14	74.62	55.18	75.07	12.18	745.30
2	10/16/99	157.59	75.99	55.72	74.67	12.73	817.91
3	10/17/99	157.21	78.16	55.41	74.79	12.47	784.17
4	10/18/99	157.25	78.82	56.54	75.72	12.87	815.74
5	10/19/99	157.33	77.97	58.04	75.49	12.84	892.37

4.3 Uncertainty Analysis

The procedure that is used to estimate the error in determining the fouling resistance is discussed and outlined in this section.

Assuming Y to be the dependent variable and X_i to be the independent variables (i = 1,.....,n), the error (ΔY) in calculating Y is given by [76, 77] as:

$$\Delta Y^2 = \sum_{i=1}^{n}\left[\frac{\partial Y}{\partial X_i}\right]^2 \Delta X_i^2 \tag{4.81}$$

The fouling resistance is given as follow:

$$R_f = \frac{1}{U_a} - \frac{1}{U_c} = U_a^{-1} - U_c^{-1} \tag{4.82}$$

The error in R_f can be written as:

$$\Delta^2 R_f = U_a^{-4}\,\Delta^2 U_a + U_c^{-4}\,\Delta^2 U_c \tag{4.83}$$

where the actual heat transfer coefficient U_a is given as:

$$U_a = \frac{\dot{Q}}{A\Delta T_{\ln} F} \tag{4.84}$$

And the total heat transfer rate $\dot{Q}$ is given as:

$$\dot{Q} = \dot{m}_t c_p \left(T_{t,in} - T_{t,out}\right) \tag{4.85}$$

Using equation (4.81), the error in U_a can be expressed as:

$$\Delta^2 U_a = \left(A\Delta T_{\ell n} F\right)^{-2} \Delta^2 \dot{Q} + \left(\frac{\dot{Q}}{AF\Delta T_{\ell n}}\right)^2 \Delta^2 \Delta T_{\ell n} \tag{4.86}$$

Assuming the log mean temperature $\Delta T_{\ell n} = 4\,\Delta T_{t,in}$

Since the errors $\Delta T_{t,in} = \Delta T_{t,out} = \Delta T_{s,in} = \Delta T_{s,out}$, the error in the total heat transfer rate can be written as:

$$\Delta^2 \dot{Q} = \left[C_p \left(T_{t,in} - T_{t,out}\right)\right]^2 \Delta^2 \dot{m}_t + 32\left[\dot{m}_t C_p\right]^2 \Delta^2 T_{t,in} \tag{4.87}$$

The clean overall heat transfer coefficient, U_c, is written as:

$$U_c = \left(\frac{1}{h_s} + \frac{D_o}{2K}\ln\left(\frac{D_o}{D_i}\right) + \frac{D_o}{h_t D_i}\right)^{-1} \tag{4.88}$$

Using equation (4.81), the error in U_c can be written as:

$$\Delta^2 U_c = \left(\frac{1}{h_s} + \frac{D_o}{2K}\ln\left(\frac{D_o}{D_i}\right) + \frac{D_o}{h_t D_i}\right)^{-4} \left[h_s^{-4}\,\Delta^2 h_s + \frac{D_o}{D_i} h_t^{-4}\,\Delta^2 h_t\right] \tag{4.89}$$

The tube side heat transfer coefficient, h_t, is given as:

$$h_t = 0.23\frac{k_t}{D_i}\mathrm{Re}_t^{0.8}\,\mathrm{Pr}_t^{0.4} \tag{4.90}$$

Using equation (4.81), the error in h_t is given as:

$$\Delta^2 h_t = \left[0.023\frac{k_t}{D_i}(0.8)\mathrm{Re}_t^{-0.2}\,\mathrm{Pr}_t^{0.4}\right]^2 \Delta^2\mathrm{Re}_t \tag{4.91}$$

where the tube side Reynolds number is written by:

$$\mathrm{Re}_t = \frac{\dot{m}_t D_i}{\mathrm{A}\mu_t}$$

where

$$\dot{m}_t = \rho_t V_t \mathrm{A}$$

And

$$\mathrm{A} = N_T \frac{\pi}{4} D_i^2$$

Hence, the error in Re_t can be written as:

$$\Delta\mathrm{Re}_t = \frac{4}{N_T \pi D_i}\Delta\dot{m}_t \tag{4.92}$$

In order to determine the error in the shell side heat transfer coefficient Δh_s the type of analysis (i.e. BD or FS) should be identified.

BD analysis method:

The shell side heat transfer coefficient is given by:

$$h_s = h_c J_c J_L J_B \tag{4.93}$$

Using equation (4.81), the error in h_s is given as:

$$\Delta h_s = J_c\, J_L\, J_B\, \Delta h_c \tag{4.94}$$

where:

$$h_c = \frac{k_s}{D_o}\left(0.211 \mathrm{Re}_s^{0.651} \mathrm{Pr}_s^{0.34}\right) \tag{4.95}$$

And the error in h_c is given as:

$$\Delta^2 h_c = \left[0.211 \frac{k_s}{D_o}(0.651)\mathrm{Re}_s^{-0.349} \mathrm{Pr}_s^{0.34}\right]^2 \Delta^2 \mathrm{Re}_s \tag{4.96}$$

And the error in Reynolds number is given as:

$$\Delta \mathrm{Re}_s = \frac{\rho_s D_o}{\mu_s} \Delta V_{\max} \tag{4.97}$$

where V_{max} is the maximum velocity and the error in V_{max} is given as:

$$\Delta V_{\max} = \frac{\Delta \dot{m}_s}{\rho_s S_m} \tag{4.98}$$

<u>*FS analysis method:*</u>

Similarly the error in the shell side heat transfer coefficient is given as:

$$\Delta^2 h_s = \left[0.211 \frac{k_s}{D_o}(0.651)\mathrm{Re}^{-0.348} \mathrm{Pr}_s^{0.34}\right] \Delta^2 \mathrm{Re} \tag{4.99}$$

while the error in Reynolds number, Re, is given as:

$$\Delta \mathrm{Re} = \frac{D_o F_{cv}}{\mu_s S_m} \Delta \dot{m}_s \tag{4.100}$$

$\Delta \dot{m}_s$ is the error in the shell side flow measurements.

The above described error estimation procedure for thermal analysis method is illustrated through the following example:

Consider the data given in table A2 and the instruments' inaccuracies presented in section 4.2:

$\Delta \dot{m}_t = \Delta \dot{m}_s = 1.5\%$

$\Delta T_{t,out} = \Delta T_{t,in} = \Delta T_{s,out} = \Delta T_{s,in} = \Delta = 1.5\%$

It can be shown that the percentage error (due to instrumentations) in R_f using both thermal analysis methods will be about ±8.5%.

4.4 Results and Discussion

The resluts of thermal analysis for both exchangers case 1 and case 2 are given in the following sections:

4.4.1 Exchanger Case 1

The construction, process, and thermal data for exchanger case 1 are shown in Table A1. Also, samples of the operation data (flow rates and temperatures for both shell and tube sides) are shown in Table 4.2 and Table A2 for one fouling cycle. A fouling cycle is defined as the period from the time when the exchanger was put in operation as new or clean to the time when the exchanger was shut down for cleaning as a result of severe fouling (i.e. when the exchanger reached a critical level of fouling and can not perform the intended heat capacity). Exchanger case 1 has 8 fouling cycles during two years of operations.

The first set of results pertains to the variation of mass flow rates and temperatures during a fouling cycle. From experimental point of view, we prefer to maintain the inlet temperatures and flow rate. However, due to operation constraints such as naphtha demand, there are variations of about 30% in the flow rate and about 15% in the inlet temperatures. Figure 4.8 shows the variation of mass flow rates with time for one of the fouling cycles. It could be seen that the shell side flow rate is higher than the tube side flow rate by about 2 kg/s and the variation of each flow rate with time is always less than 3 kg/s. Figure 4.9 shows the temperatures variations for the same fouling cycle. As can be seen, there are no sharp variation (more than 8° C) of all temperatures with time.

The second set of results pertains to the variation of overall heat transfer coefficients with time. The Bell Delaware (BD) method described in section 4.1 is used to evaluate the heat transfer coefficient for the shell side. Samples of the results for exchanger case 1 using BD method is shown in Table 4.4 and Table A3.

Table 4.4: Samples of the Results Using BD Method for Exchanger Case 1

Day	Re_s	h_c ($W/m^2 K$)	J_C	J_L	J_B	h_s ($W/m^2 K$)	Re_t
1	6564.2	929.62	1.02	0.86	0.93	758.44	9823.69
3	7458.5	1010.23	1.02	0.86	0.93	824.21	10679.41
5	7398.3	1004.91	1.02	0.86	0.93	819.87	10937.63
7	8030.9	1060.04	1.02	0.86	0.93	864.85	10938.33
9	7991.6	1056.66	1.02	0.86	0.93	862.09	10937.90

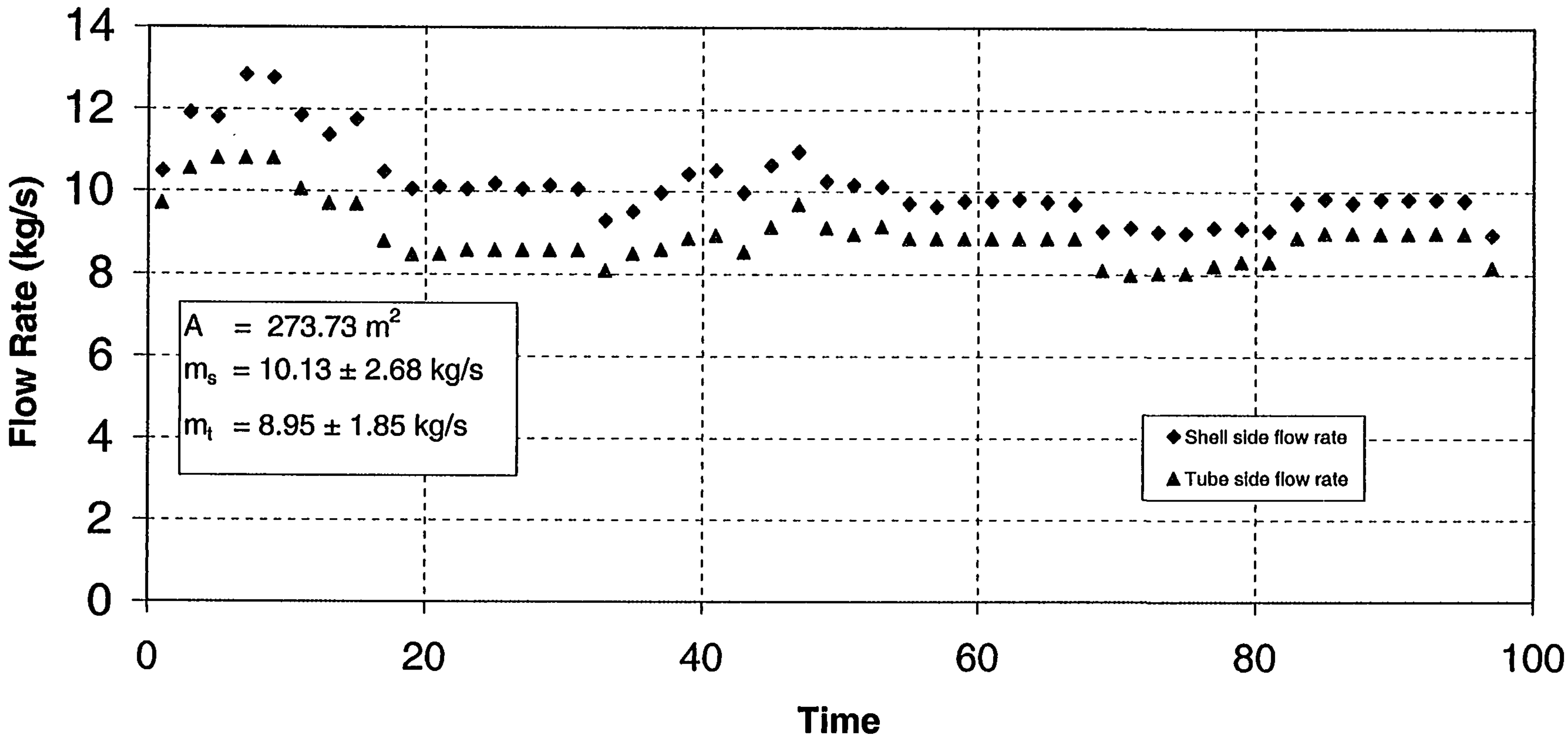


Figure 4.8: Tube and Shell-Side Flow Rates Versus Time for Exchanger Case 1

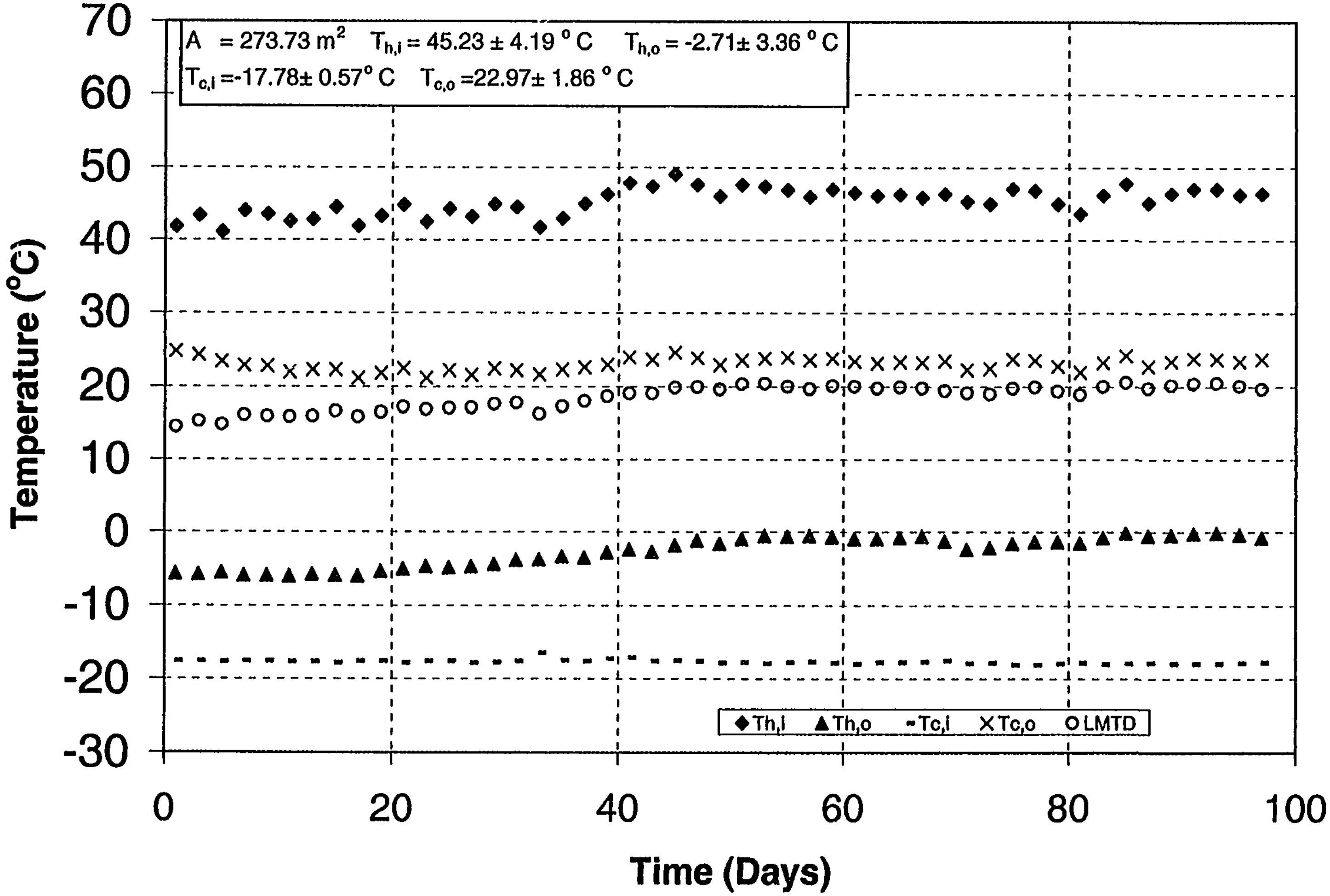


Figure 4.9: Operating Temperatures Versus Time for Exchanger Case 1

Table 4.4: Samples of the Results Using BD Method for Exchanger Case 1 (cont.)

Day	h_t (W/m² K)	R_k (W/m² K)	U_c (W/m² K)	LMTD	F	U_a (W/m² K)	Fouling Resistance (hr. ft². F/BTU)
1	792.16	5.01E-05	334.47	14.37	0.78	287.13	0.00280
3	846.90	5.01E-05	359.74	15.18	0.80	297.88	0.00328
5	863.24	5.01E-05	362.65	14.72	0.81	297.04	0.00346
7	863.29	5.01E-05	371.20	16.02	0.83	283.35	0.00475
9	863.26	5.01E-05	370.68	15.83	0.83	284.97	0.00461

The change of cross flow and the corrected heat transfer coefficients as a function of time for the shell side during the same fouling cycle, is shown in Figure 4.10. The variation in the heat transfer coefficients with time during the same fouling cycle exhibits the same trend as the variation of the flow rates. These coefficients are the theoretical values and are not functions of fouling growth. The corrected heat transfer coefficient (h) is less than the cross flow heat transfer coefficient (h_c) by a factor J (less than unity), which accounts for baffle configuration (J_c), leakage (J_L), and bypass (J_B). These factors depend on the heat exchanger geometrical construction. An initial guess value of 0.6 is recommended in the literature [25]. A value of 0.81 is calculated for the geometry under study. The large difference between the value reported in the literature and the calculated one indicates the importance of determining this factor (J) for each specific geometry. Based on the calculated values of heat transfer coefficients, log mean temperature difference, temperature correction factor and heat balance on one of the flow streams, the actual and the clean (theoretical) heat transfer coefficient as function of time are calculated and

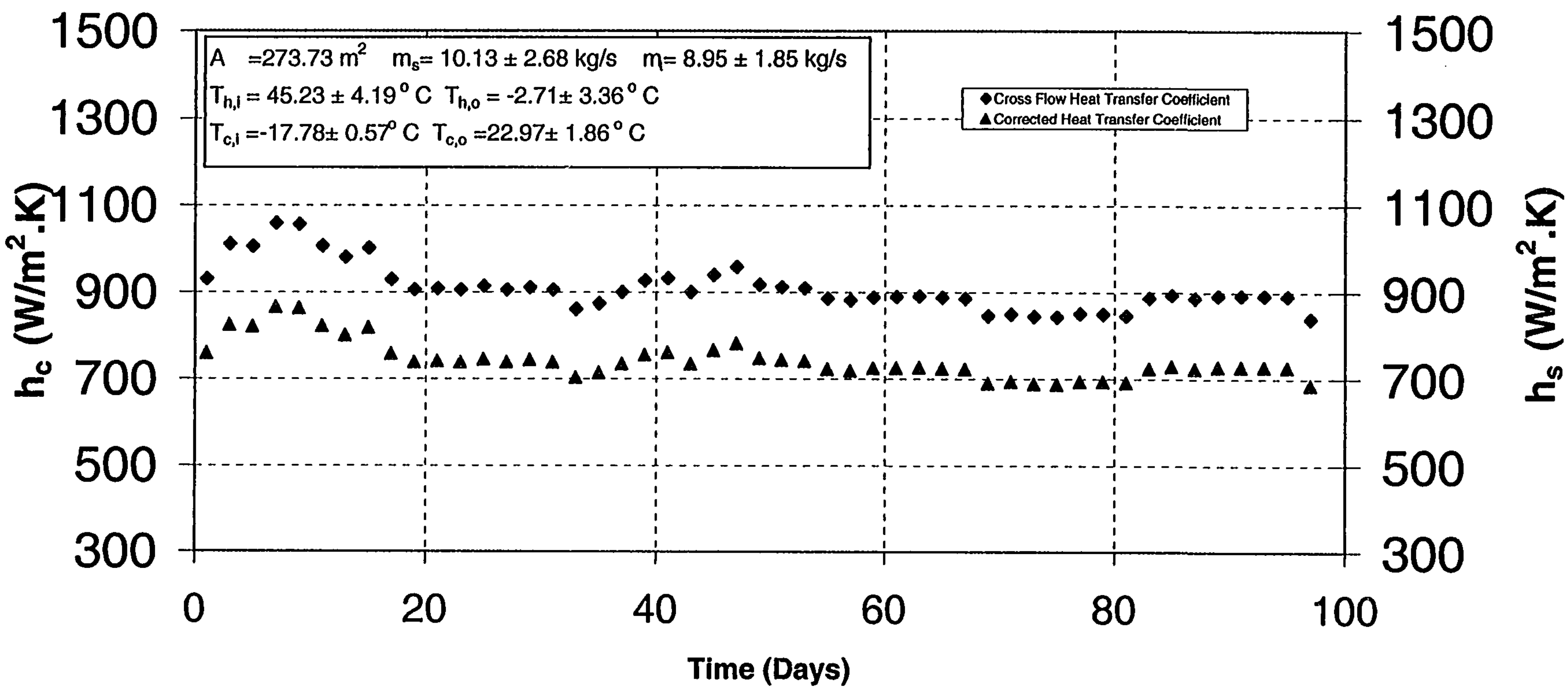


Figure 4.10: Cross Flow and Corrected Shell-Side Heat Transfer Coefficients Versus Time for Exchanger Case 1

shown in Figure 4.11. It can be observed that the actual overall heat transfer coefficient decreases with time while the clean overall heat transfer coefficient exhibit the same trend as the mass flow rates. The reduction of the actual overall heat transfer coefficient with time is a clear indication of fouling.

The third set of results pertains to the variation of fouling growth with time during the same fouling cycle. Figure 4.12 shows the variation of fouling growth with time. The figure indicates that the fouling is increasing with time for the subject heat exchanger. The fouling resistance, in this work, is represented in English units since it is commonly used by heat exchanger designers, manufactures and industry. Industrial standard [78, 79, 80] define the clean service as the one in which the fouling is less than or equal to 0.002 hr.ft^2.F/Btu (0.000352 m^2 C/W). From Figure 4.12, it is clear that the subject heat exchanger has experienced severe fouling. For the fouling cycle under consideration, the fouling resistance has increased from 0.002 hr.ft^2.F/Btu (0.000352 m^2 C/W) to 0.015 hr.ft^2.F/Btu (0.00264 m^2 C/W) in 4 months period. Similar trends are obtained for different fouling cycles for the subject heat exchanger as shown in Figures 4.13a, 4.13b and 4.13c. These fouling cycles have different time duration. The plots exhibit a power law trend and it will be used in the statistical analysis. Figure 4.13a (fouling cycle 2) indicates that the fouling rate of growth (the slope which is not the fouling growth) decreases with time. This is attributed to the decrease of the deposition rate and the increase of fouling removal rate as a result of the increase in the flow rate and thus the flow velocity. Some calculated parameters are related to fouling and

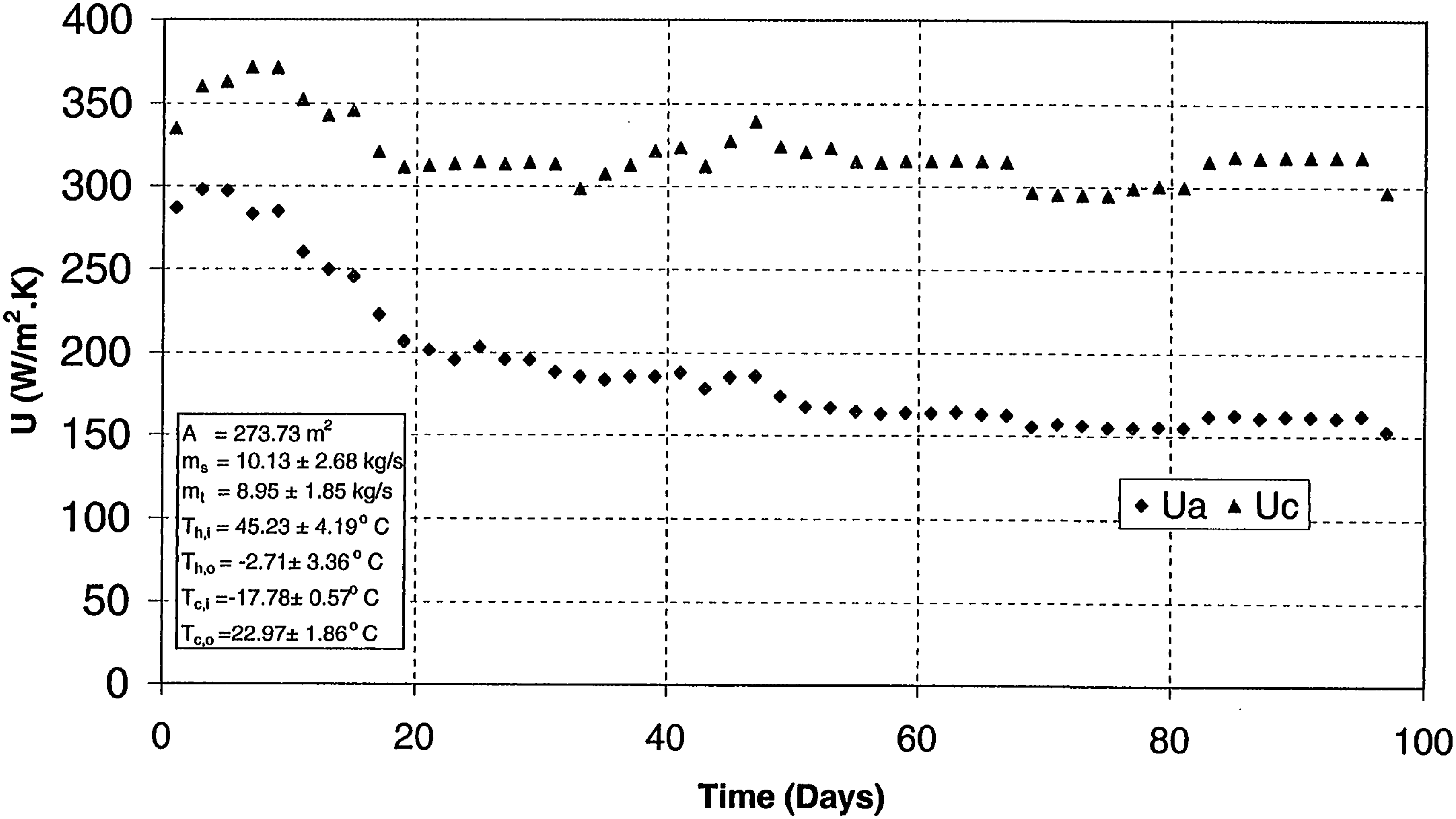


Figure 4.11: Actual and Clean Overall Heat Transfer Coefficients Versus Time for Exchanger Case 1

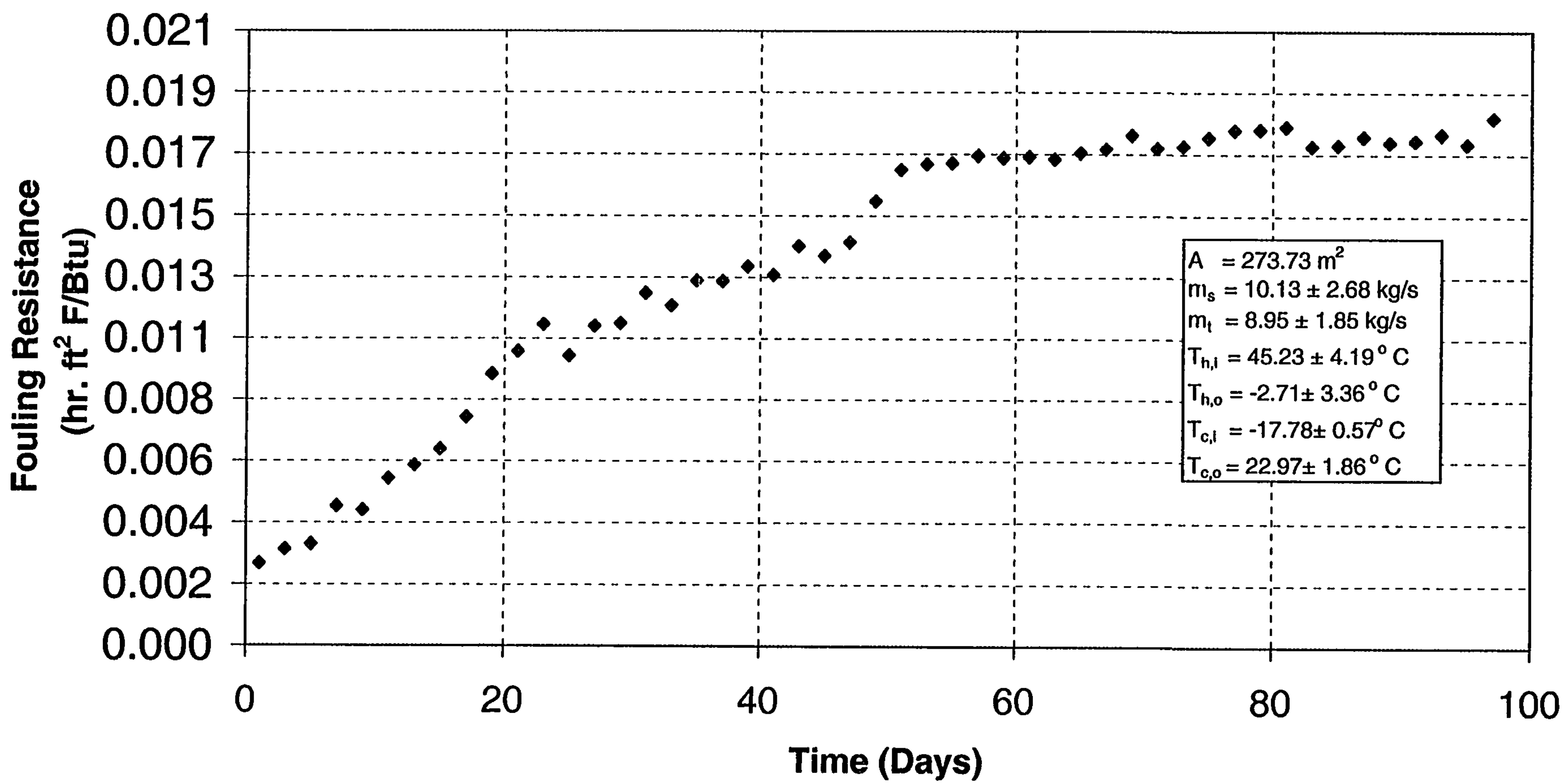


Figure 4.12: Fouling Growth Versus Time for Exchanger Case 1 Using BD Method

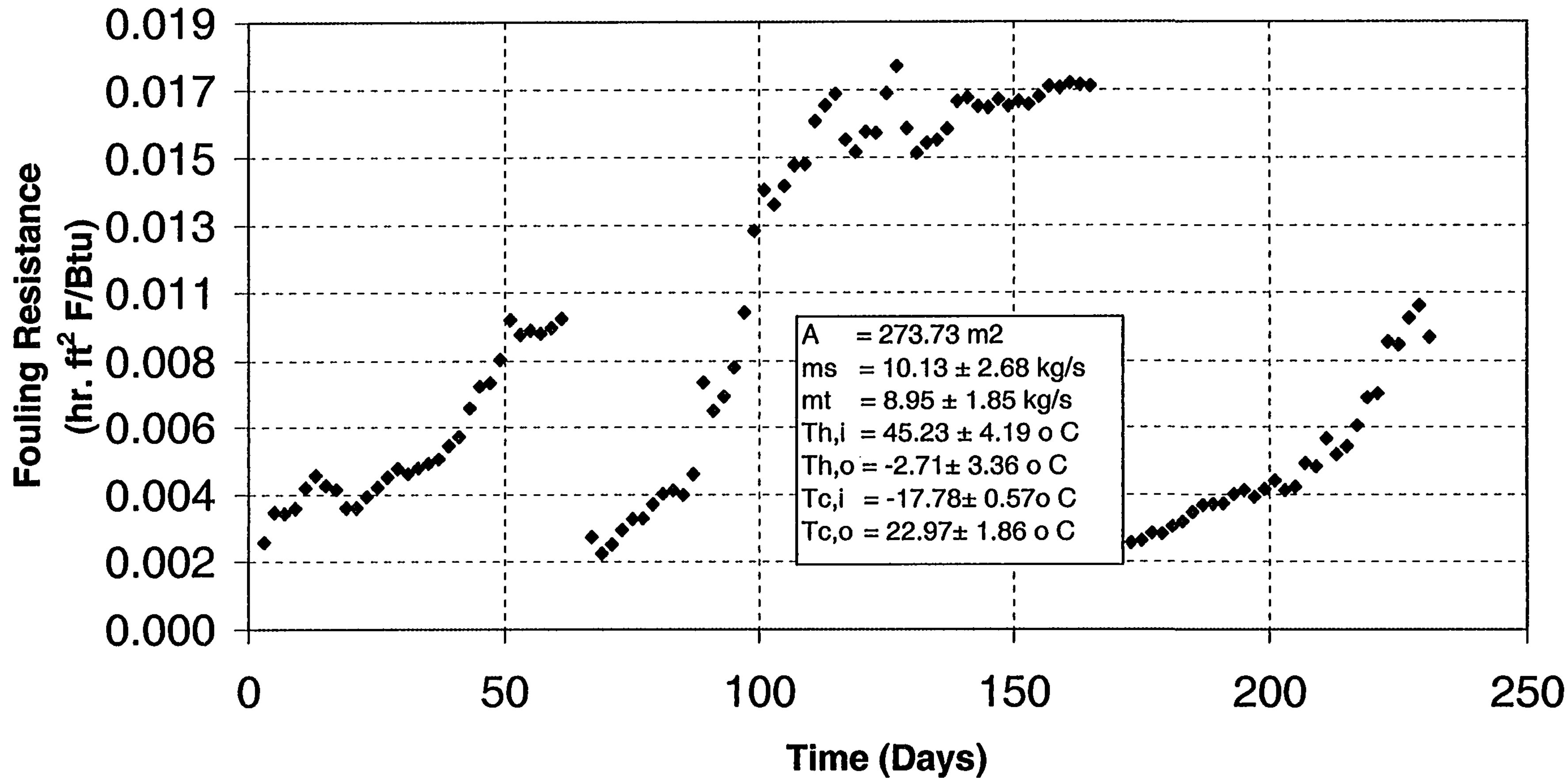


Figure 4.13a: Fouling Growth Versus Time for Fouling Cycles 1, 2, and 3 Using BD Method for Exchanger Case 1

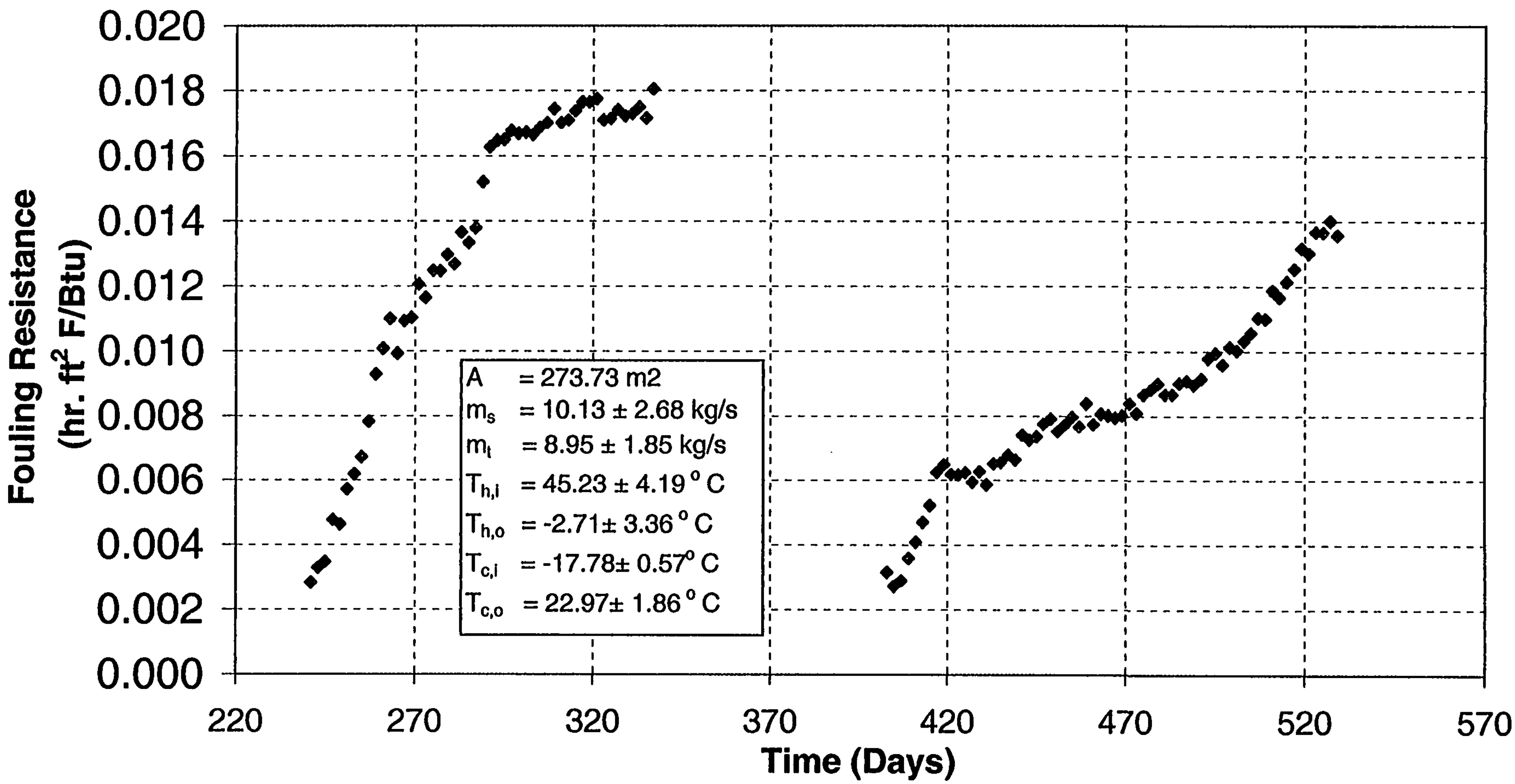


Figure 4.13b: Fouling Growth Versus Time for Fouling Cycles 4, and 5 Using BD Method for Exchanger Case 1

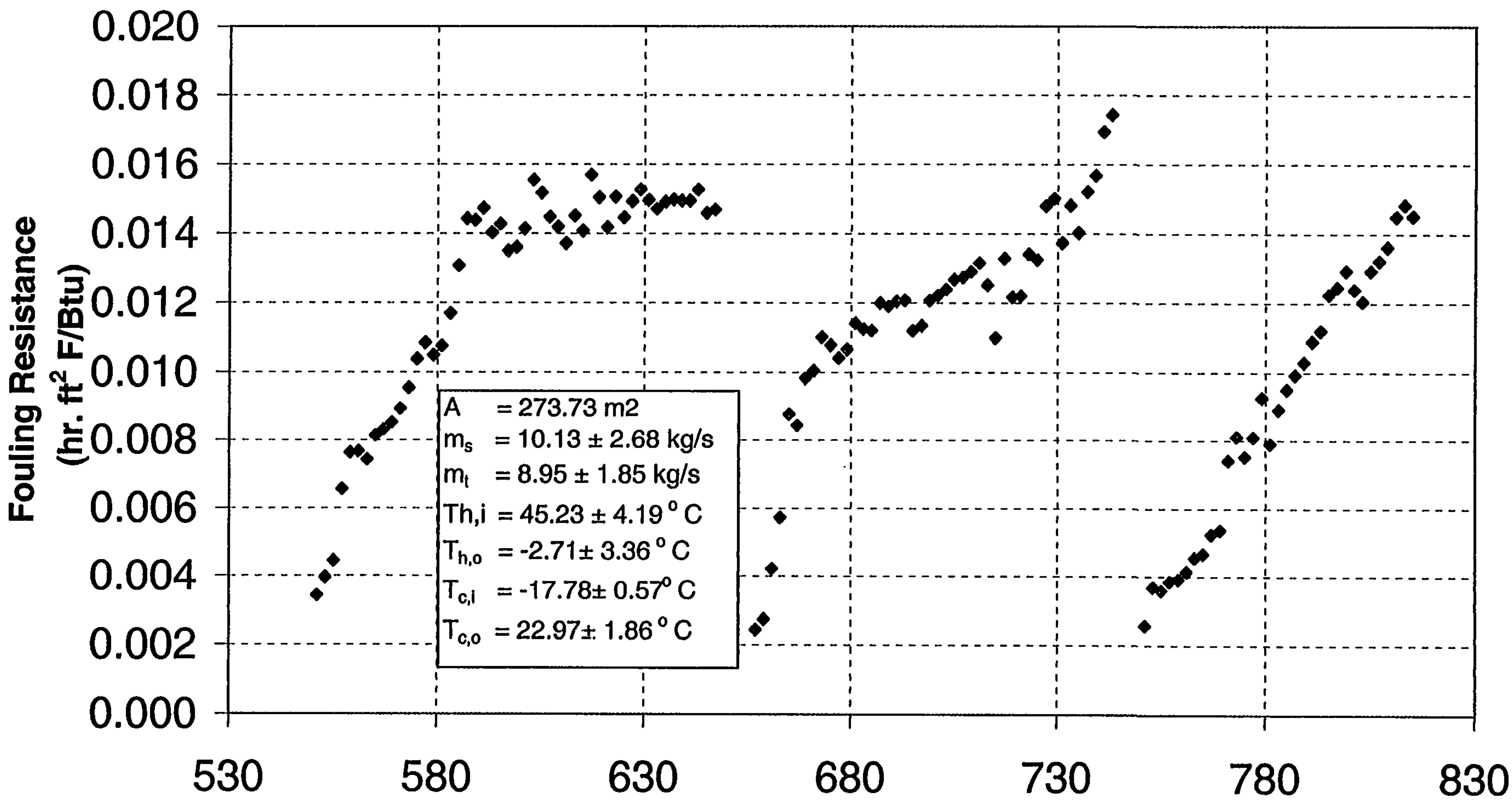


Figure 4.13c: Fouling Growth Versus Time for Fouling Cycles 6, 7, and 8 Using BD Method for Exchanger Case 1

could be used as an indication of thermal degradation but they will not give quantitative measure of fouling. Such parameters include the actual overall heat transfer coefficient and the corrected log mean temperature difference. A plot of U_{act} and corrected LMTD against the fouling resistance (Figure 4.14) indicates that the decrease in the actual overall heat transfer coefficient and the increase in the corrected log mean temperature difference are an indication of increase in fouling resistance.

The forth set of results pertains to the calculation of fouling growth using flow stream analysis (FSA) method. Using the mathematical procedure described in section 4.1, the parameters needed to determine the fouling resistance are calculated. Samples of the calculated results are shown in Table 4.5 and Table A.4.

Table 4.5: Samples of the Results Using FS Method for Exchanger Case 1

Day	Date	Re_s	F_b	F_s	F_t	h_s (W/m^2 K)	Re_t
1	4/19/99	3318.02	0.15	0.13	0.21	596.23	9823.69
3	4/21/99	3784.89	0.15	0.13	0.21	649.58	10679.41
5	4/23/99	3753.35	0.15	0.13	0.21	646.05	10937.63
7	4/25/99	4085.53	0.15	0.13	0.21	682.72	10938.33
9	4/27/99	4064.81	0.15	0.13	0.21	680.46	10937.90

Table 4.5: Samples of the Results Using FS Method for Exchanger Case 1 (cont.)

Day	R_k (W/m^{2}K)	h_t (W/m^2 K)	U_c (W/m^2 K)	LMTD	F	U_a (W/m^{2}K)	Fouling Resistance (hr. ft^2.F/BTU)
1	5.01E-05	792.16	302.55	14.37	0.78	287.13	0.00101
3	5.01E-05	846.90	326.51	15.18	0.80	297.88	0.00167
5	5.01E-05	863.24	328.69	14.72	0.81	297.04	0.00184
7	5.01E-05	863.29	337.93	16.02	0.83	283.35	0.00324
9	5.01E-05	863.26	337.38	15.83	0.83	284.97	0.00310

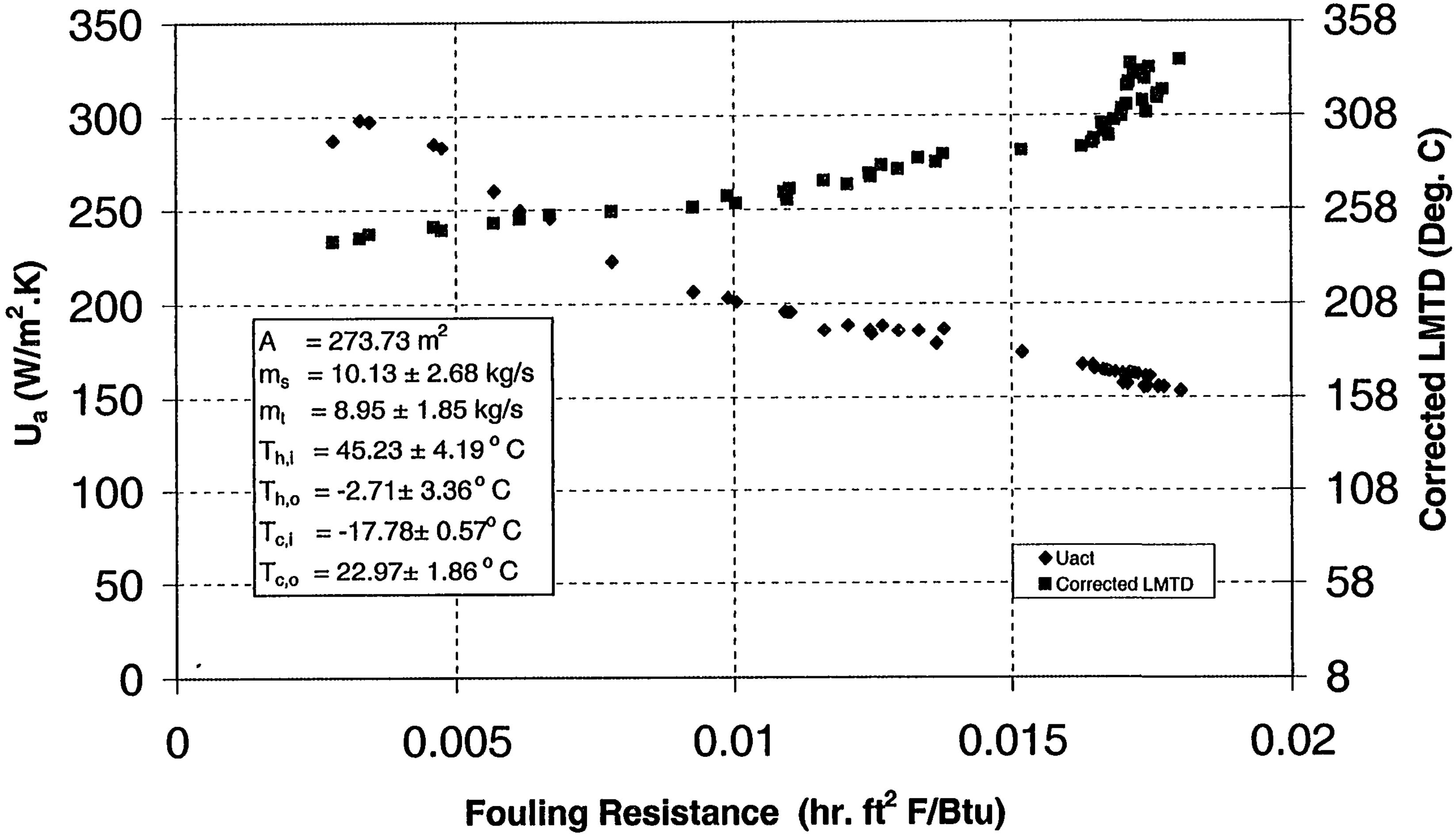


Figure 4.14: Actual Overall Heat Transfer Coefficient and Corrected LMTD Versus Fouling Growth for Exchanger Case 1

The plot of the actual and clean overall heat transfer coefficients (using FSA method) versus time, for the same fouling cycle considered earlier using Bell Delaware method, is shown in Figure 4.15. As can be seen, the actual overall heat transfer coefficient decreases with time while the clean overall heat transfer coefficient remains almost constant. This is expected since the actual overall heat transfer coefficient is a function of heat transfer rate, inlet and outlet temperatures, temperature correction factor, and heat transfer area while the clean overall heat transfer coefficient is a function of exchanger geometry and flow rates, which do not change during the fouling cycle. Figure 4.16 shows the plot of fouling growth versus time. As can be seen and expected, fouling increases with time. A similar trend is observed for the other fouling cycles as shown in Figures4.17a, 4.17b, and 4.17c. Figure 4.16 shows a comparison for the fouling growth observed using BD and FS analysis methods. The fouling resistances obtained using BD method are higher than those obtained using FS analysis method. In order to determine the accuracy of each method, the t test and F test are used. If the mean, variance, and standard deviation are represented by μ_1, σ_1^2, and σ_1 for the BD method and by μ_2, σ_2^2, and σ_2 for the FS analysis method, respectively, the statistical hypothesis that both methods give the same results is written as:

$$H_o: \quad \mu_1 = \mu_2$$

whereas the alternative hypothesis that they give different results are written as:

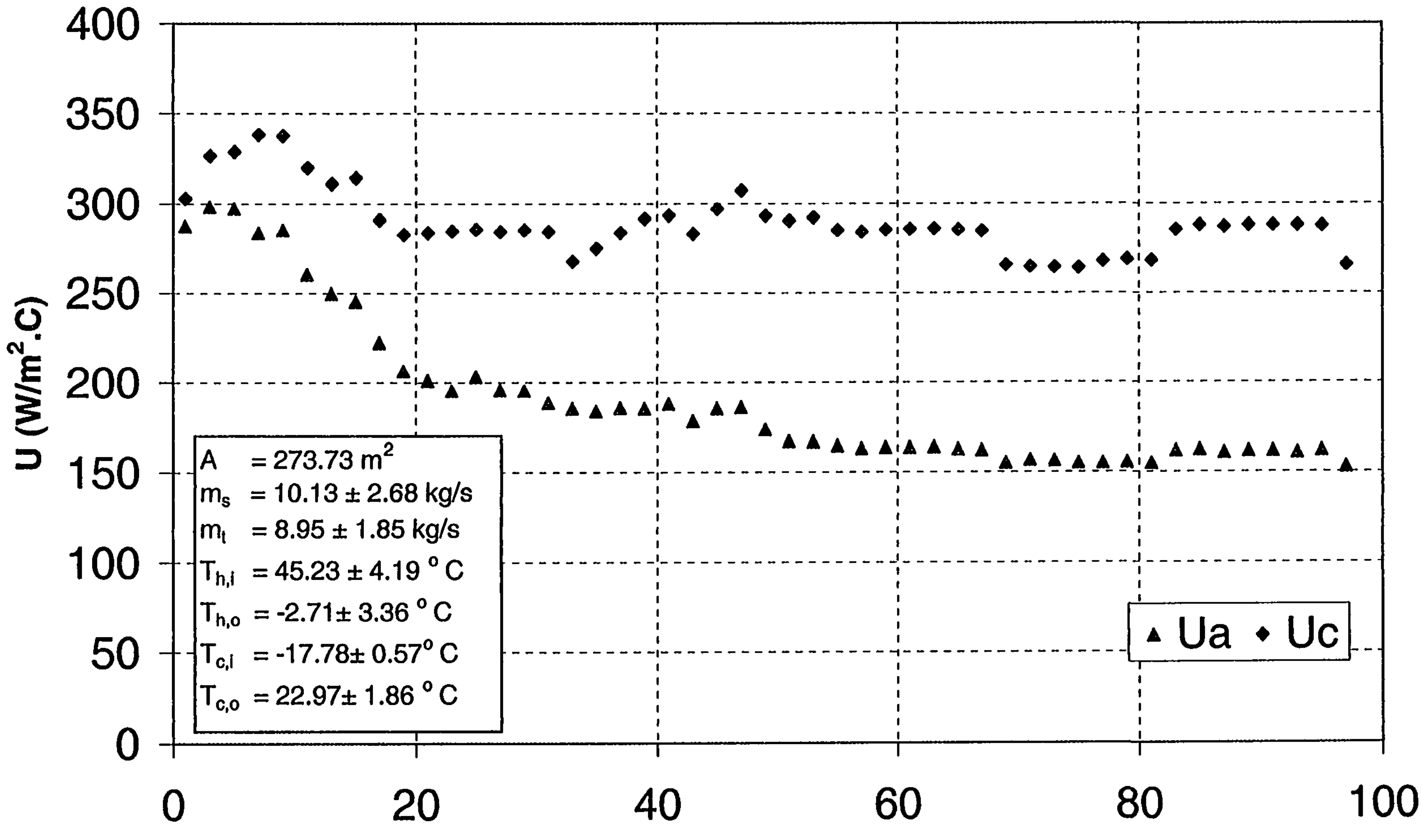


Figure 4.15: Actual and Clean Overall Heat Transfer Coefficients Versus Time for Exchanger Case 1 Using FS Method

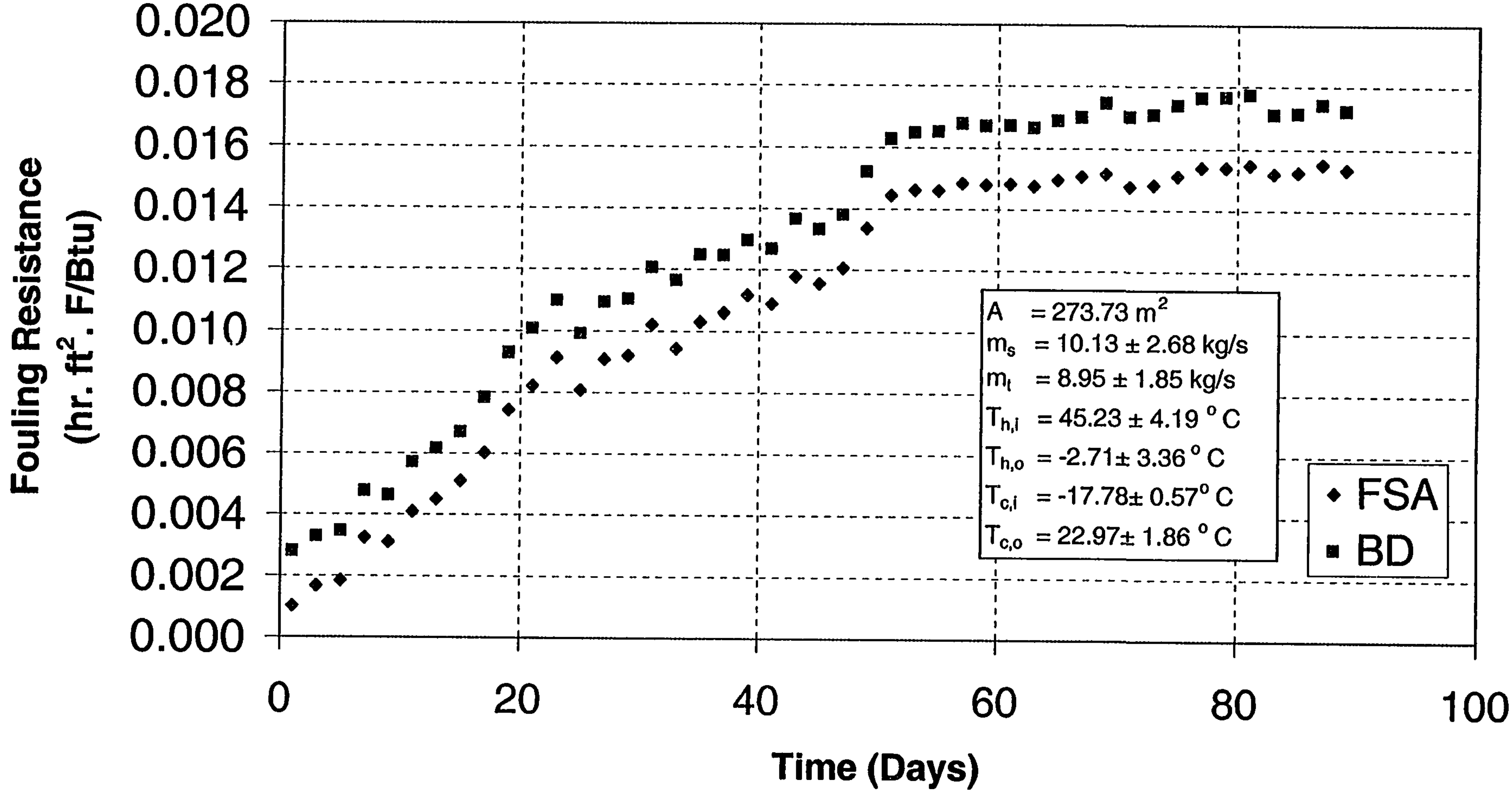


Figure 4.16: Comparison of Fouling Growth Versus Time for Exchanger Case 1 Using BD and FSA Methods

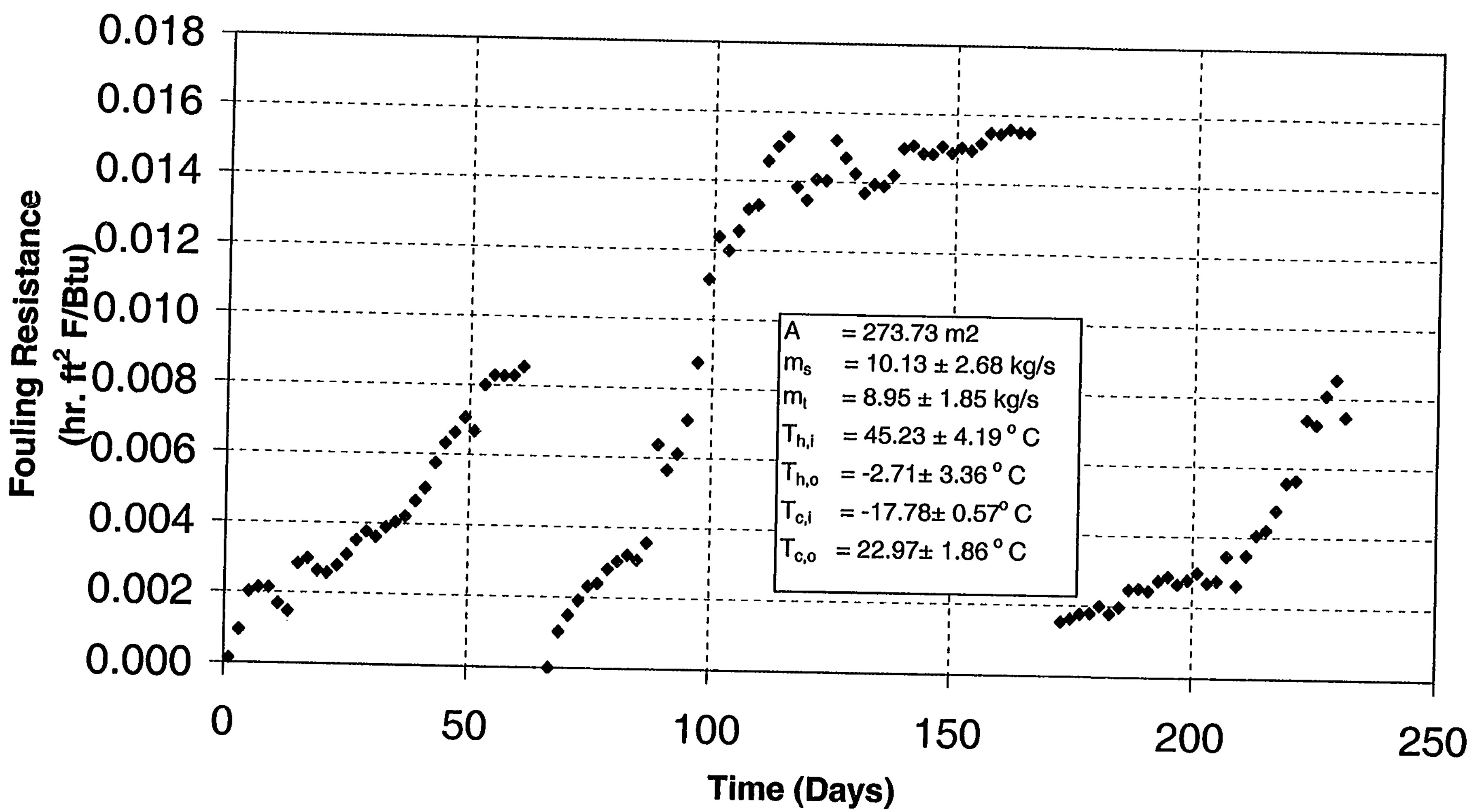


Figure 4.17a: Fouling Growth Versus Time for Fouling Cycles 1, 2, and 3 Using FS Method for Exchanger Case 1

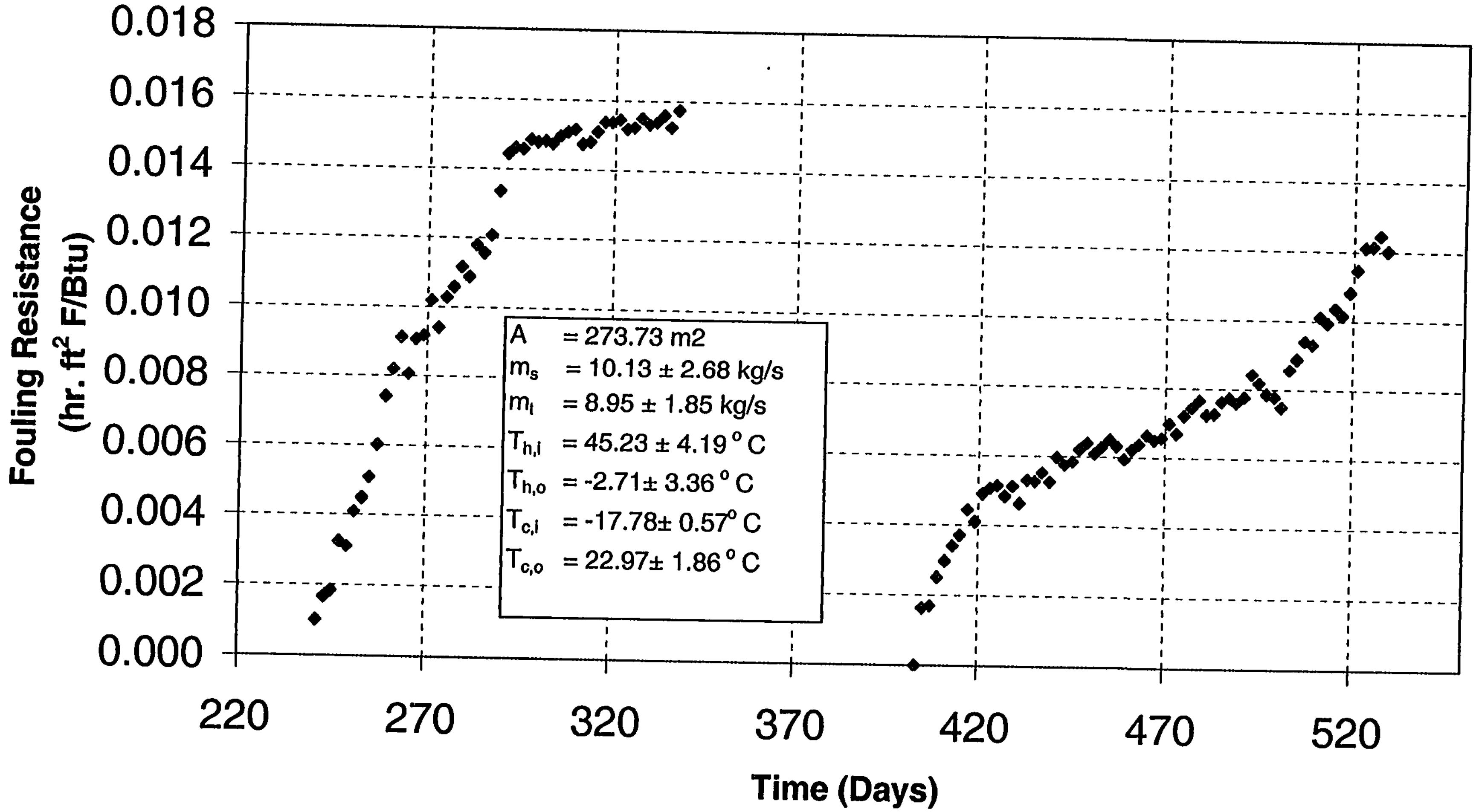


Figure 4.17b: Fouling Growth Versus Time for Fouling Cycles 4, and 5 Using FS Method for Exchanger Case 1

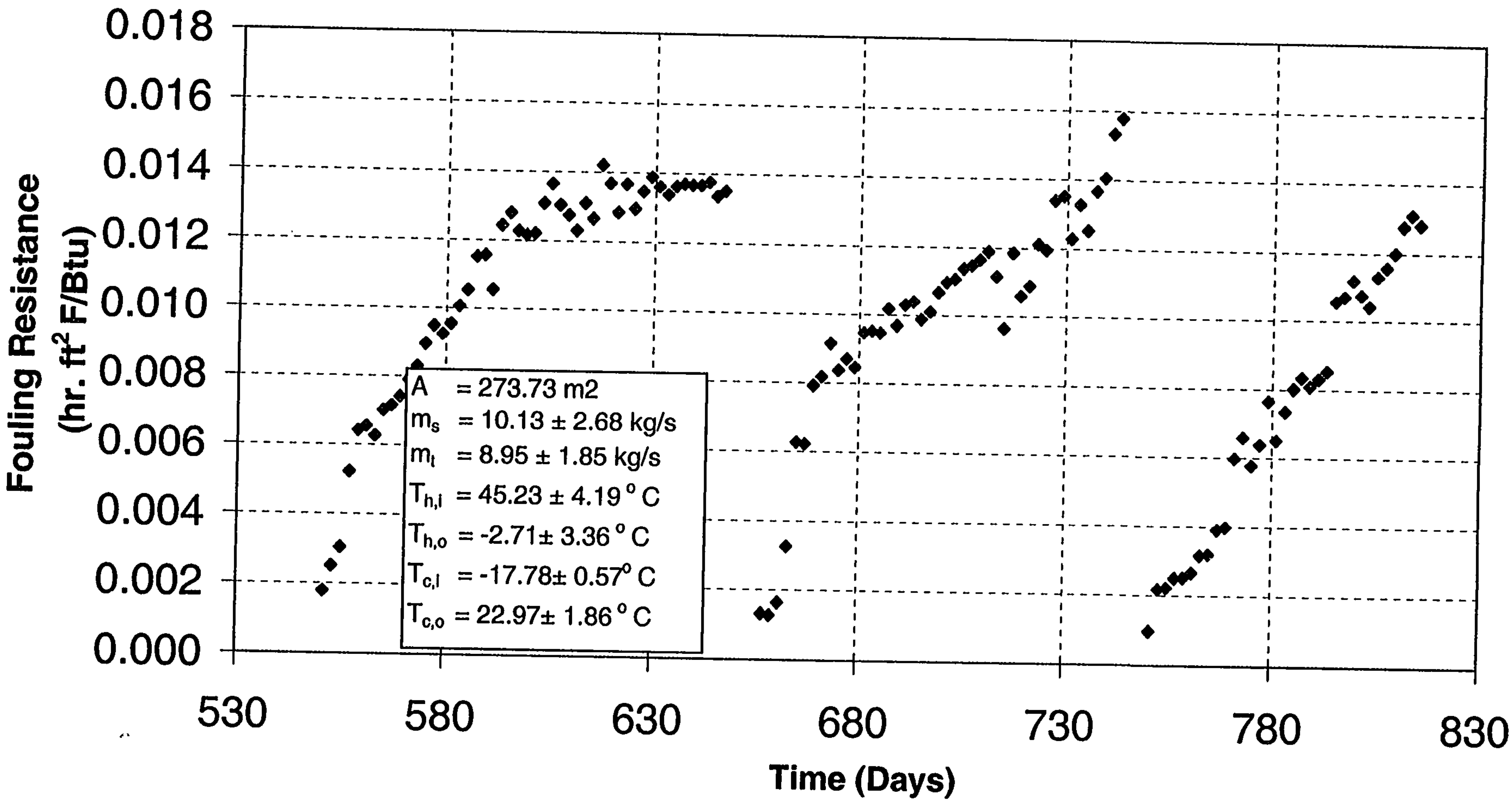


Figure 4.17c: Fouling Growth Versus Time for Fouling Cycles 6, 7, and 8 Using FS Method for Exchanger Case 1

$H_a: \quad \mu_1 \neq \mu_2$

To decide which t test to be used, a test of equality of variances is conducted first. If we accept the equality of variances, a t test with a pooled variance (S_p^2) will be used. Otherwise, an alternative t test will be used. The hypothesis that both methods have same variance is given as:

$H_o: \quad \sigma_1^2 = \sigma_2^2$

and the alternative hypothesis that both methods have different variances is written as:

$H_a: \quad \sigma_1^2 \neq \sigma_2^2$

The test statistic F_o is calculated as:

$$F_o = \frac{S_1^2}{S_2^2}$$

The hypothesis H_o will be rejected if $F_o > F_{\alpha/2, n_1-1, n_2-1}$.

To test a hypothesis a procedure has to be devised for taking a random sample, computing an appropriate test statistic, and then rejecting or failing to reject the hypothesis H_o. Part of this procedure is specifying the significant level of the test α, (type 1 error) which is the probability of rejecting H_o given that H_o is true. A value of α = 0.05 will be used in this analysis as recommended in the literature [81]. The value of test statistic t_o for the equal means hypothesis ($H_o: \ \mu_1 = \ \mu_2$) is given by:

$$t_o = \frac{\hat{\mu}_1 - \hat{\mu}_2}{S_p \sqrt{\frac{1}{n_1} + \frac{1}{n_2}}}$$

Where n_1 and n_2 are the number of sample points for BD and FS, respectively.

An estimate of the common variance S_p^2 is given by:

$$S_p^2 = \frac{(n_1 - 1)\sigma_1^2 + (n_2 - 1)\sigma_2^2}{n_1 + n_2 - 2}$$

The hypothesis H_o is rejected if $|t_o| > t_{\alpha/2, n_1+n_2-2}$ where $t_{\alpha/2, n_1+n_2-2}$ is the upper $\alpha/2$ percentage point of the t distribution with $n_1 + n_2 - 2$ degrees of freedom. If H_o is rejected, it can be concluded fouling growth results obtained using BD and FSA methods do differ.

Using the above analysis, a summary of the results is given in Table 4.6. Since $|t_o| > t_{\alpha/2, n_1+n_2-2}$, the hypothesis H_o is rejected and hence the fouling growth results obtained using BD and FSA methods do differ. Also, since $F_o > F_{\alpha/2, n_1-1, n_2-1}$, both methods have same variance.

Table 4.6: Comparison Between BD and FS Analysis Methods

	BD	FS
Mean (μ)	0.01322	0.01132
Variance (σ^2)	2.19704×10^{-5}	2.0682×10^{-5}
Number of sample points (n)	49	49
Common standard deviation (S_p)	0.004618	
Test statistic (F_o)	1.062	
Test statistic (t_o)	2.039	
$F_{\alpha/2,n_1-1,n_2-1}$	1.880	
$t_{\alpha/2,n_1+n_2-2}$	1.988	
Conclusion	• Fouling results form BD and FS methods is different. • The variability between the two methods is the same.	

Since the comparison of the two thermal methods (i.e. BD and FS) indicates that they are different, then which one is more accurate? In order to answer this question both results are compared with results obtained using a proprietary thermal package program called STX (Shell and Tube Software for Process Industry) [82]. Usually the source code for such programs is not accessible, however, the results obtained from these programs are very reliable and accurate. Several contactors and designers use STX since 1976. Thousands of heat exchangers designed by STX were built, installed, and then tested in a range of services covering nearly all petrochemical/refineries applications. The results confirm that this program is very reliable. Table 4.7a shows fouling growth results obtained using the STX program, BD and FS analyses methods. Also, Table 4.7b shows the shell side heat transfer coefficients obtained using the

STX program, BD and FS analyses methods. The results from STX program are shown in Appendix C.

Table 4.7a : Fouling Factor Results from STX Program, BD, and FS

Results	STX	BD	FS
1	0.01410	0.01205	0.01019
2	0.01850	0.01627	0.01443
3	0.01906	0.01685	0.01495
4	0.01936	0.01708	0.01518
5	0.02026	0.01803	0.01573

Table 4.7b : Shell side Heat Transfer Coefficients from STX Program, BD, and FS (W /m^2 K)

Results	STX	BD	FS
1	967.80	737.96	579.67
2	973.91	743.00	583.76
3	950.88	723.90	568.37
4	949.84	723.11	567.68
5	905.48	684.60	524.00

The t test results for both thermal analysis methods in comparison with STX program are shown in Tables 4.8a and 4.8b.

Table 4.8a : Comparison Between BD and STX Results

	BD	STX
Mean (μ)	0.01606	0.01826
Variance (σ^2)	$5.41999\text{x}10^{-6}$	$5.803\text{x}10^{-6}$
Number of sample point (n)	5	5
Common standard deviation (S_p)	0.00205	
Test statistics (t_o)	1.693	
$t_{\alpha/2,n_1+n_2-2}$	2.306	
Conclusion	Fouling results from BD and STX programs are same.	

Table 4.8b : Comparison Between FS and STX Results

	FS	STX
Mean (μ)	0.0141	0.018256
Variance (σ^2)	$4.984 \text{ x } 10^{-6}$	$5.803\text{x}10^{-6}$
Number of sample point (n)	5	5
Common standard deviation (S_p)	0.0020515	
Test statistics (t_o)	3.267	
$t_{\alpha/2,n_1+n_2-2}$	2.306	
Conclusion	Fouling results from FS and STX programs are different.	

The results indicate that the Bell Delaware method is more accurate than the FSA method since the BD method gave results that are in close agreement with the ones obtained using the STX program. The thermodynamics properties are considered constant in the present study. To justify such assumption, the effect of the variable thermodynamics properties on the overall heat transfer coefficient (U_c) is studied based on the methods presented by Shah [26] and Thomas [83]. The results of such study are given in Table 4.9. The comparison between the overall heat transfer coefficients (with constant thermodynamics properties) used in the present study and those obtained using Shah [26] method, indicates almost negligible difference, while the comparison with the values obtained using Thomas [83] method, indicates a maximum difference of 2.6%. Hence, the use of constant thermodynamics properties in the present study is justified.

Table 4.9: Comparisons of the Overall Heat Transfer Coefficient for Constant and Variable Thermodynamics Properties

Data #1	U_c with Constant Properties	Professional Version Software (Thomas Method)		Shah Method	
		U_c	% Difference	U_c	% Difference
1	268.06	271.154	1.141	268.13	0.027
2	323.56	332.379	2.653	323.64	0.027
3	338.19	339.803	0.472	338.29	0.028
4	373.19	382.31	2.385	373.30	0.032
5	338.59	347.65	2.605	338.70	0.034

4.4.2 Exchanger Case 2

The discussion in this section deals with the results of fouling analysis in a second type of heat exchanger (case 2) where the tube side is subjected to fouling. Geometrical and one fouling cycle operation data for the subject heat exchanger are given in Table 4.3 and Tables B1 and B2. The data indicates that the variation in shell side flow rate is less than 7% while the variation in the tube side flow rate reaches up to 45%. The variation of the inlet and outlet temperatures for one fouling cycle for both fluids versus time is shown in Figure 4.18. There is a slight variation of temperature with time. Using energy balance for each phase and the calculation procedure described in section 4.1, the parameters needed to determine the fouling rate are calculated and a sample of these results are shown in Table 4.10 and Table B3.

Table 4.10: Samples of the Thermal Results for Exchanger Case 2

Day	$LMTD_v$ (°C)	$LMTD_{pc}$ (°C)	$LMTD_l$ (°C)	$LMTD_{wtd}$ (°C)	$\%\dot{Q}_v$	$\%\dot{Q}_{pc}$	$\%\dot{Q}_l$
1	79.02	85.03	47.18	75.21	0.39	83.37	16.24
2	80.14	85.08	47.81	75.55	0.52	83.34	16.14
3	79.83	85.12	50.04	76.42	0.48	83.32	16.19
4	78.92	84.12	49.30	75.54	0.49	83.48	16.03
5	79.19	83.69	46.87	74.42	0.50	83.68	15.82

Table 4.10: Samples of the Thermal Results for Exchanger Case 2 (cont.)

Day	A_v (m^2)	A_{pc} (m^2)	A_l (m^2)	L_{pc} (m)	U_c (W/m^2 K)	U_a (W/m^2 K)	Fouling Resistance (hr. ft². F/BTU)
1	2.09	513.56	180.53	2.21	652.56	588.08	0.000955
2	2.67	518.82	174.70	2.24	693.31	612.00	0.001089
3	2.53	525.17	168.49	2.26	675.29	593.14	0.001165
4	2.58	523.75	169.86	2.26	691.46	618.00	0.000977
5	2.45	526.58	167.15	2.27	734.18	624.28	0.001363

It is worth mentioning that the hot fluid experiences a phase change when it cools down from superheated to subcooled liquid phase. The results indicate that the amount of heat transfer rate during the phase change represents above 83% of the total amount of heat transfer rate. 16% of the total amount of heat transfer rate takes place during the subcooled region while the remaining 1% takes place in the superheated region. Based on this, the thermal analysis is performed for each phase.

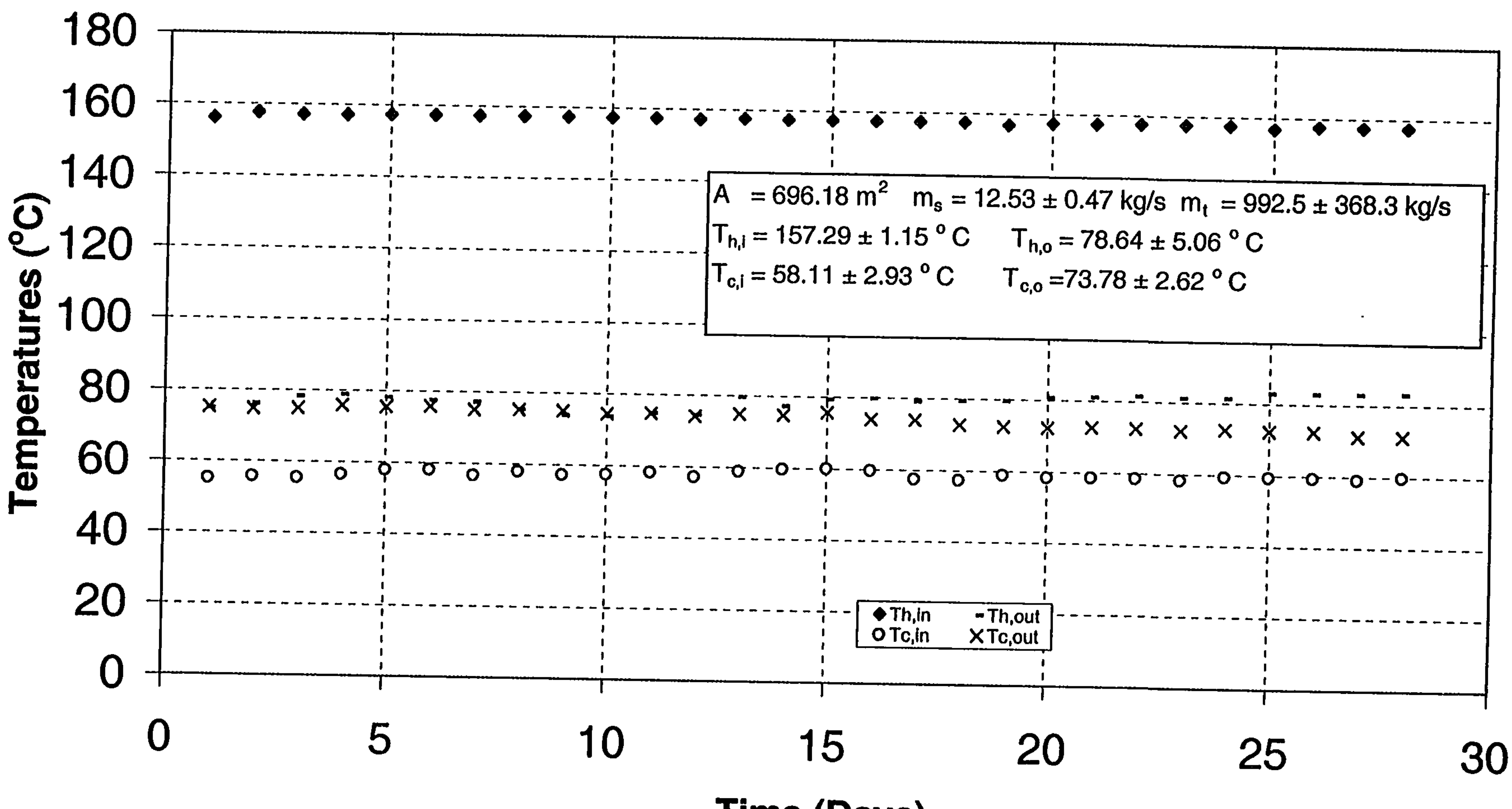


Figure 4.18: Temperatures Versus Time for One Fouling Cycle for Exchanger Case 2

The first set of results for the subject heat exchanger (case 2) pertains to the overall heat transfer coefficients using BD method. Figure 4.19 shows the plot of the clean overall heat transfer coefficients for each phase versus time. From the figure, it could be seen that the overall heat transfer coefficients increase with time. This is because the tube side flow rate is increased by increasing the level of the crude in the column in order to compensate for the fouling growth. This is done intentionally by increasing the crude level in the column above the reboiler inlet to increase the tube side circulation and thus compensate for degradation due to the fouling.

The second set of results pertains to the products of the clean overall heat transfer coefficient and the associated area for each phase as shown in Figure 4.20. From the figure, it is clear that the significant contribution is made by the two phase region. The contribution due to the liquid phase is small while the contribution due to the superheated phase is negligible.

The third set of results pertains to the fouling growth. The fouling growth versus time for eleven fouling cycles are shown in Figures 4.21a, 4.21b, 4.21c and 4.21d. For the fouling cycle under consideration, the fouling growth is shown in Figure 4.22. For this specific fouling cycle, fouling increased from a value of 0.0001 hr.ft^2.F/Btu (0.0000176 m^2 C/W) to a value of 0.0035 hr.ft^2.F/Btu (0.000616 m^2 C/W) in about 28 days. Similar trends that are exhibited by heat exchanger case 1 are observed for heat exchanger case 2.

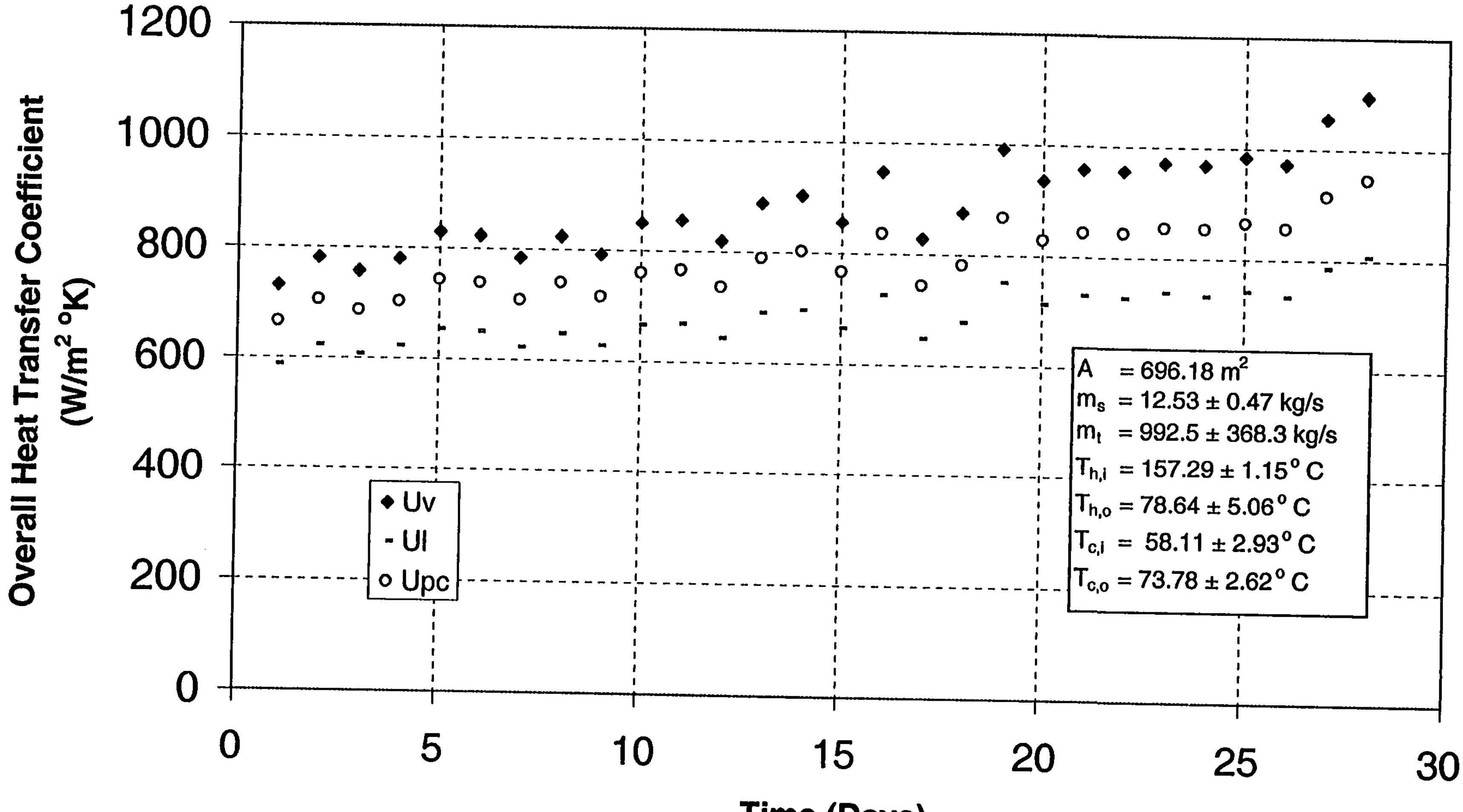


Figure 4.19: Overall Heat Transfer Coefficient for Each Phase Versus Time for One Fouling Cycle for Exchanger Case 2

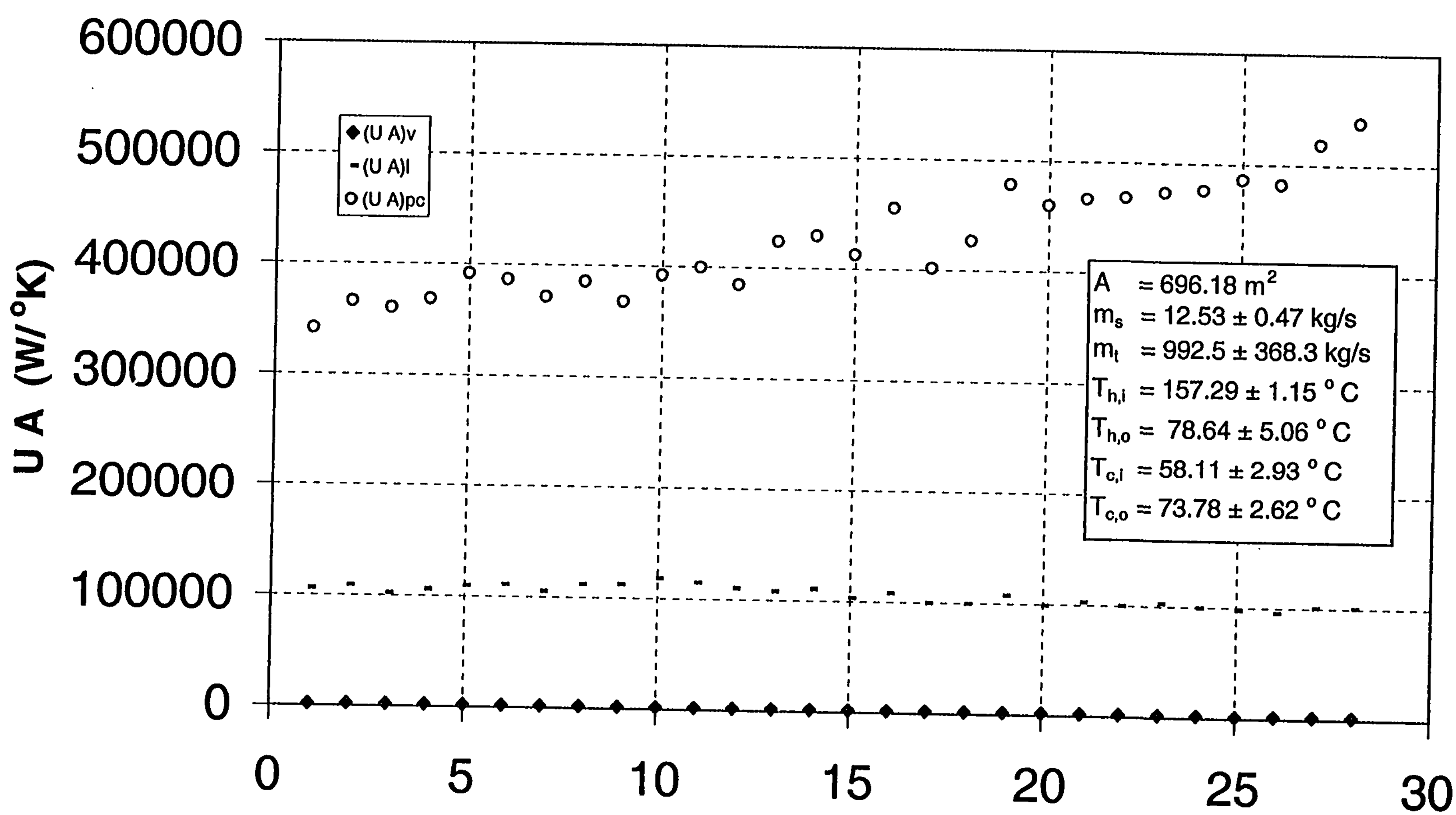


Figure 4.20: Product of Area and Overall Heat Transfer Coefficient for Each Phase Versus Time for One Fouling Cycle for Exchanger Case 2

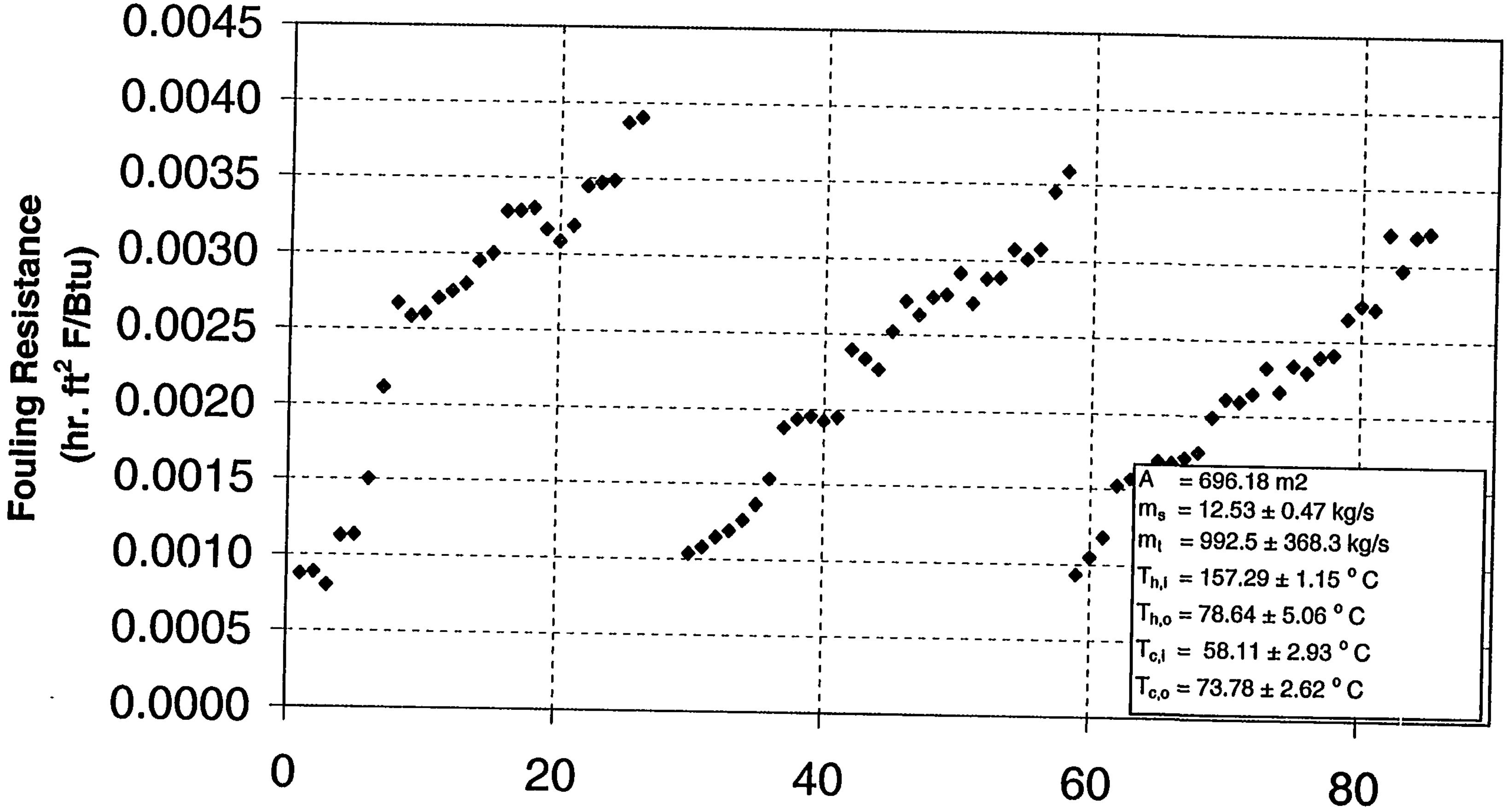


Figure 4.21a: Fouling Growth Versus Time for the First Three Fouling Cycles (Cycles 1, 2, and 3) for Exchanger Case 2

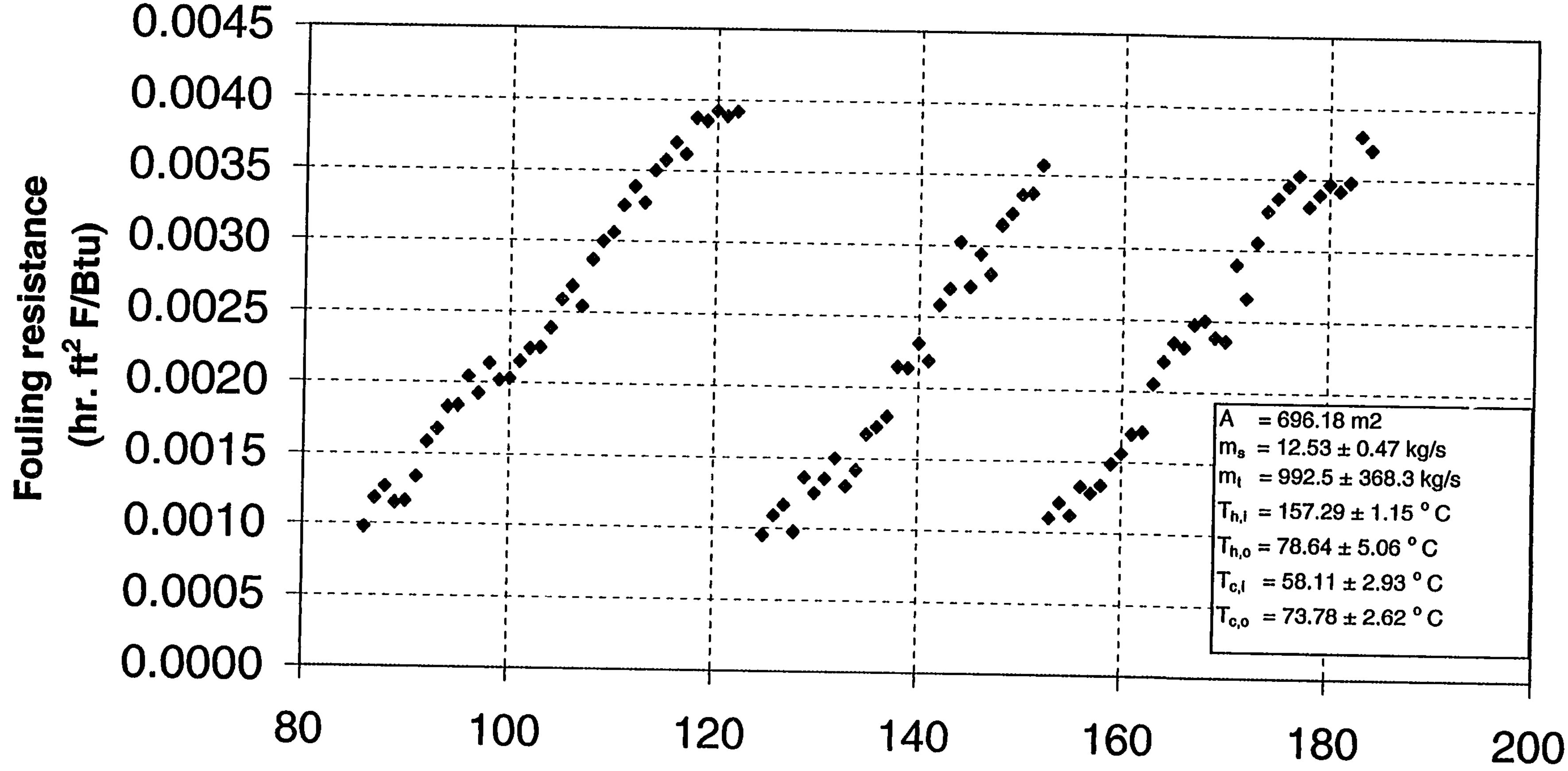


Figure 4.21b :Fouling Growth Versus Time for the Second Three Fouling Cycles (Cycles 4, 5, and 6) for Exchanger Case 2

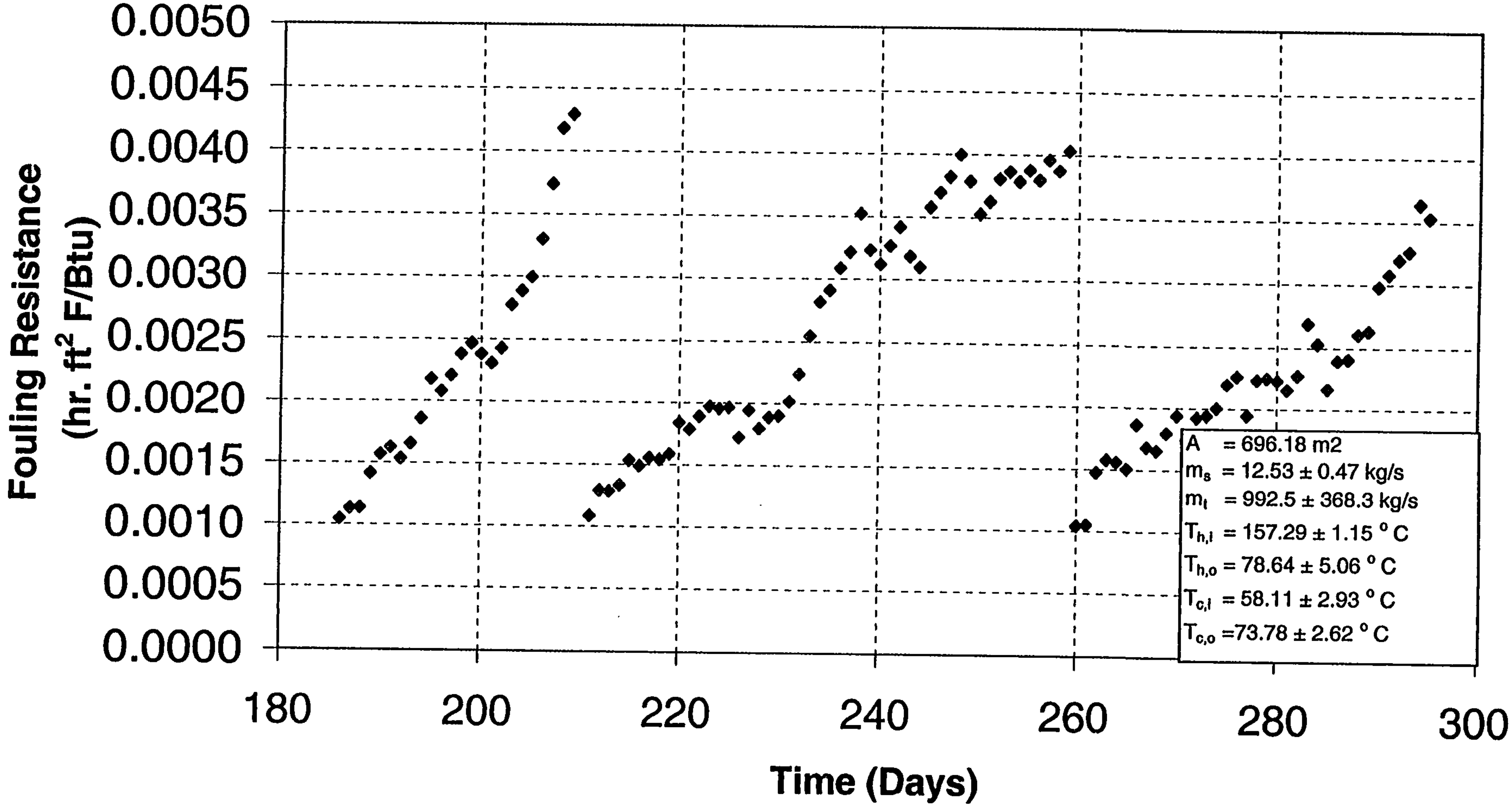


Figure 4.21c: Fouling Growth Versus Time for the Third Three Fouling Cycles (Cycles 7, 8, and 9) for Exchanger Case 2

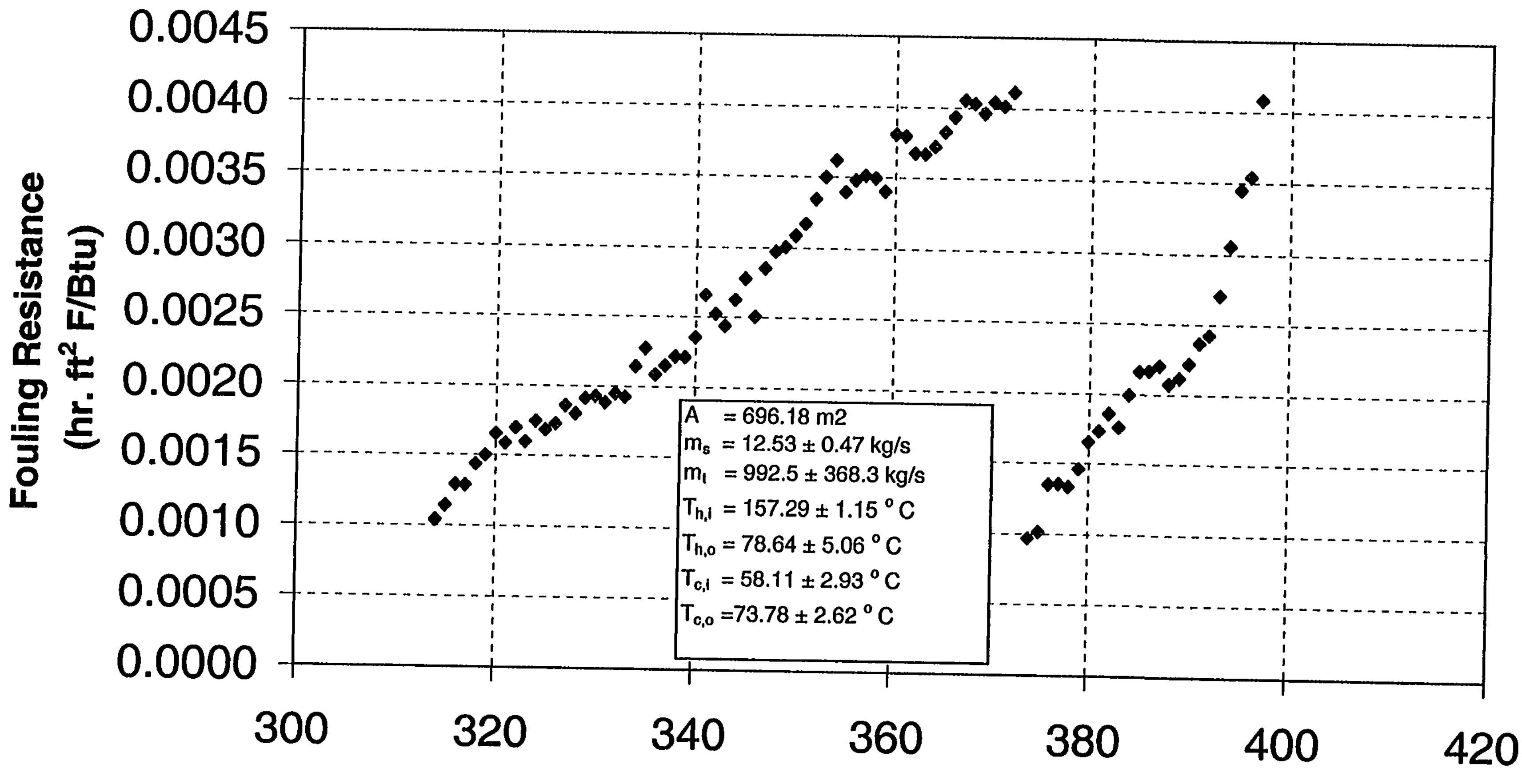


Figure 4.21d: Fouling Growth Versus Time for the Last Fouling Cycles (Cycles 10, and 11) for Exchanger Case 2

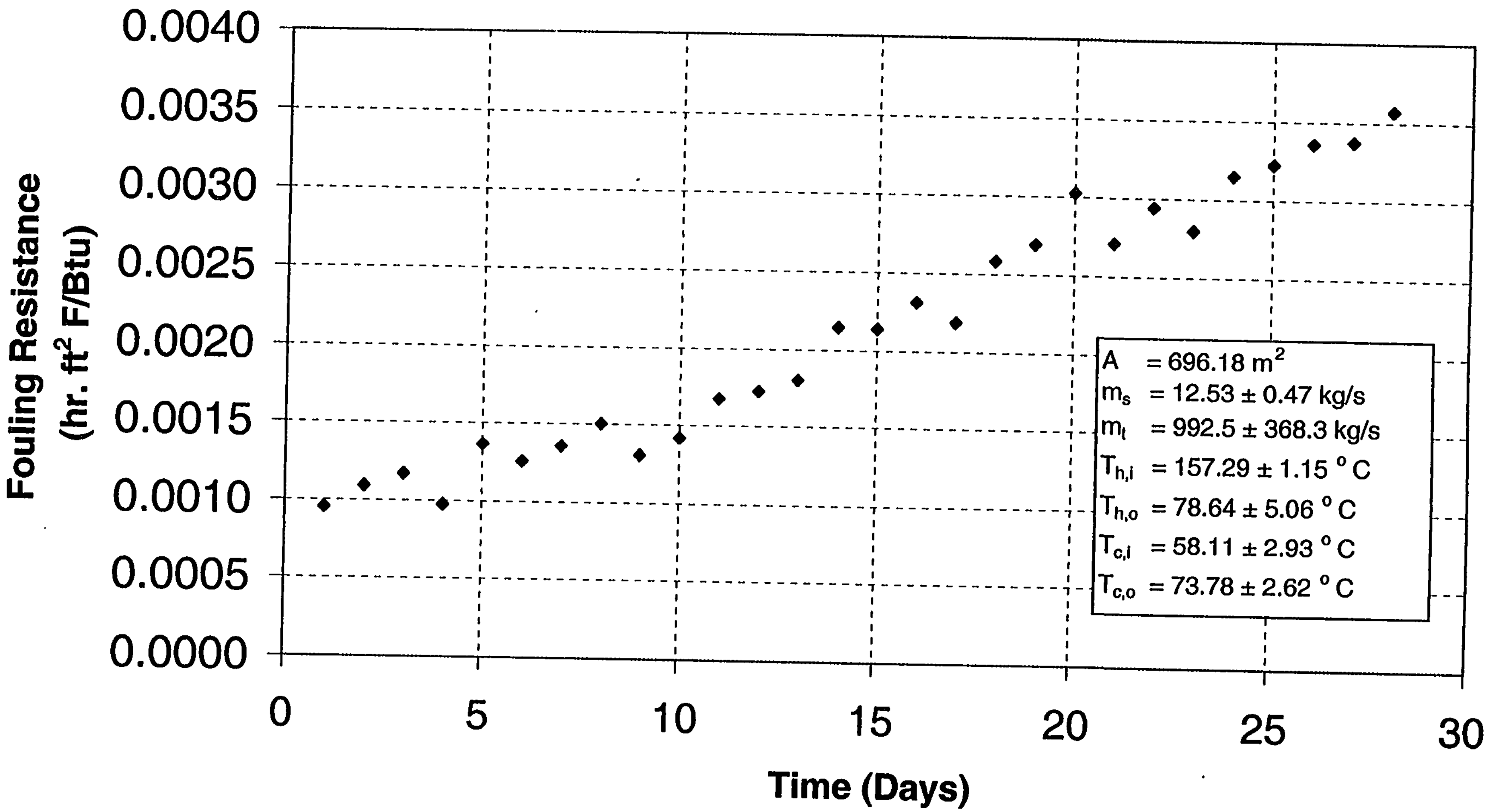


Figure 4.22: Fouling Growth Versus Time for One Fouling Cycle for Exchanger Case 2

CHAPTER 5

FOULING PREDICTION

This chapter is related to fouling prediction. It introduces a statistical approach, which is used to develop fouling growth models. This approach is applied on both exchanger case 1 and exchanger case 2. The developed fouling models could be used to schedule maintenance of the exchangers.

5.1 Statistical Analysis

Statistical analysis has been used by several researchers [6, 37, 36, 84, 85] to characterize the actual time required to reach the critical level of fouling in heat exchangers. In this study, a statistical approach to formulate a fouling model for heat exchangers is developed. The approach required finding the probability distributions for the fouling data of the exchanger under consideration [86]. The fouling data are the time to reach to the critical level of fouling during a period of observation. After finding the distribution, the best distribution with the highest coefficient of determination or best K-S statistics will be utilized to develop the fouling growth model. The procedure used in developing the model is detailed below:

For the time to reach to the critical fouling level (i.e., time to clean the heat exchanger), the Cumulative Distribution Function (CDF), is written by [87, 88, 89] as:

$$F(t_i)=\frac{i}{N_f+1} \tag{5.1}$$

i= 1, ..., N_f for the i^{th} event of time to reach to the critical level of fouling, where $F(t_i)$ is cumulative value of F(t) at $t=t_i$.

The reliability function R(t), (the probability of trouble free operation of heat exchanger)

$$R(t) = 1-F(t) \tag{5.2}$$

And the cumulative hazard function H(t) is given by [87, 88, 89] as:

$$H(t) = -\ell n R(t) \tag{5.3}$$

The cumulative hazard function represents a degree of cumulative damage (in this case to the heat exchanger). If it has been decided that preventive action is needed at a specified reliability (for example 0.95%), then it implies that there is a corresponding cumulative hazard function limit when such an action need to be taken. Increasing rate of H(t) indicates that the heat exchanger surfaces are undergoing ageing type of deterioration due to fouling deposits.

Several distributions could be postulated to describe the time to critical level of fouling in heat exchangers. The collected data can be utilized for probability plotting, parameters estimation, and distribution model's validation.

An extremely useful technique for such an analysis consists of transforming the equation for the CDF to a form that can be plotted as

$$Y = aX + b \tag{5.4}$$

Therefore, the parameters for each proposed distribution could be found by the above transformation. The summary of the several distributions and their transformation as well as the relationship of their parameters with the slope "a" and intercept "b" of the regression model is given in Table 5.1. The coefficient of determination (R^2) is used to judge how well the specified model characterizes the data. The coefficient of determination (R^2) is given as:

$$R^2 = \frac{\left(\overline{XY} - \overline{X}\,\overline{Y}\right)^2}{\left(\overline{X^2} - \overline{X}^2\right)\left(\overline{Y^2} - \overline{Y}^2\right)} \tag{5.5}$$

where; $\overline{X} = \frac{1}{N}\sum_{i=1}^{N} X_i$; $\overline{Y} = \frac{1}{N}\sum_{i=1}^{N} Y_i$

$$\overline{XY} = \frac{1}{N}\sum_{i=1}^{N} \overline{X_i}\,\overline{Y_i}$$

$$\overline{X^2} = \frac{1}{N}\sum_{i=1}^{N} X_i^2 ; \overline{Y^2} = \frac{1}{N}\sum_{i=1}^{N} Y_i^2$$

Table 5.1. Distribution Models and their Transformation: Y and X, where $F(t_i) = i/(N+1)$ at $t = t_i$ [36, 87, 88, 89]

Model	Probability density function, f(t)	CDF, F(t)	X	Y	Parameters
Exponential distribution	$\lambda e^{-\lambda t} = \frac{1}{\theta} e^{-t/\theta}$	$1 - e^{-\lambda t}$	t	$\ln\left[\frac{1}{1-F(t)}\right]$	$\lambda = \frac{1}{\theta} = a, b = 0$
Weibull distribution	$\frac{m}{\theta}\left(\frac{t}{\theta}\right)^{m-1} \exp\left[-\left(\frac{t}{\theta}\right)^{m}\right]$	$1 - \exp\left[-\left(\frac{t}{\theta}\right)^{m}\right]$	ln(t)	$\ln\left\{\ln\left[\frac{1}{1-F(t)}\right]\right\}$	m = a $\theta = \exp[-(b/a)]$
Normal distribution	$\frac{1}{\sqrt{2\pi}\sigma}\exp\left[\frac{-1}{2}\left(\frac{t-\mu c}{\sigma}\right)^{2}\right]$	$\int_{-\infty}^{t} f(t)dt = \Phi\left(\frac{t-\mu c}{\sigma}\right)$	t	$\Phi^{-1}[F(t)]$	$\mu_c = -(b/a)$ $\sigma = 1/a$
Lognormal Distribution	$\frac{1}{\sqrt{2\pi}\,\omega t}\exp\left\{-\frac{1}{2\omega^{2}}\ln\left[\frac{t}{t_o}\right]^{2}\right\}$	$\Phi\left[\frac{1}{\omega}\ell n\left(\frac{t}{t_o}\right)\right]$	ln(t)	$\Phi^{-1}[F(t)]$	$t_o = \exp[-(b/a)]$ $\omega = 1/a$

The mean μ_c, the variance σ^2, and the confidence limits for these distributions are given as follow:

Exponential Distribution:

$$\mu_c = \frac{1}{\lambda} = \theta$$

$$\sigma^2 = \frac{1}{\lambda^2} = \theta^2$$

$$\theta^{\pm} = \hat{\theta} \exp\left[\pm 1.018 \left|Z_{\alpha/2}\right| N^{-1/2}\right] \tag{5.6}$$

where $|Z_{\alpha/2}|$ is as follows:

$$\left|Z_{\alpha/2}\right| = 1.28 \textit{ for } 80\% \textit{ confidence limit}$$
$$\left|Z_{\alpha/2}\right| = 1.648 \textit{ for } 90\% \textit{ confidence limit}$$
$$\left|Z_{\alpha/2}\right| = 1.96 \textit{ for } 95\% \textit{ confidence limit}$$
$$\left|Z_{\alpha/2}\right| = 2.58 \textit{ for } 99\% \textit{ confidence limit}$$

Weibull Distribution:

$$\mu_c = \theta \Gamma\left(1 + \frac{1}{m}\right)$$

$$\sigma^2 = \theta^2 \left\{ \Gamma\left(1 + \frac{2}{m}\right) - \left[\Gamma\left(1 + \frac{1}{m}\right)\right]^2 \right\}$$

$$m^{\pm} = \hat{m} \exp\left[\pm 1.049 \left|Z_{\alpha/2}\right| N^{-1/2}\right] \tag{5.7}$$

$$\theta^{\pm} = \hat{\theta} \exp\left[\pm 1.018 \left|Z_{\alpha/2}\right| N^{-1/2}\right]$$

Normal Distribution:

$$\mu_c^{\pm} = \hat{\mu}_c \pm \left| Z_{\alpha/2} \right| \frac{\hat{\sigma}}{\sqrt{2N}}$$

$$\sigma^{\pm} = \hat{\sigma} \pm \left| Z_{\alpha/2} \right| \frac{\hat{\sigma}}{\sqrt{2(n-1)}} \qquad (5.8)$$

Lognormal Distribution:

$$\mu_c = t_o \exp\left(\frac{w^2}{2}\right)$$

$$\sigma^2 = t_o^2 \exp(w^2)\left[\exp(w^2) - 1\right]$$

$$t_o^{\pm} = \hat{t}_o \exp\left[\pm \left| Z_{\alpha/2} \right| \hat{\omega} N^{-1/2}\right] \qquad (5.9)$$

$$\omega^{\pm} = \hat{\omega} \pm \left| Z_{\alpha/2} \right| \frac{\hat{\omega}}{\sqrt{2(N-1)}}$$

Using the results of the best distribution with its parameters, the fouling growth model can be developed from the knowledge of end point fouling either by physically measuring it when the heat exchanger is disassembled for cleaning, or by calculating the critical level of fouling from the mathematical relationship linking heat exchanger performance with the level of fouling (thermal analysis). The second approach is used in this study. This provided a practical way of generating appropriate fouling growth curves which can provide a clue to adjust the operational parameters to enhance the average time between cleaning. There are different fouling growth models leading to different distributions. For example, for the lognormal distribution, the fouling growth model is given by [36, 90, 91] as:

$$R_f(t) = R_f(1)t^{\beta} \tag{5.10}$$

where:

- $R_f(t)$: is fouling resistance at time t (day).
- $R_f(1)$: is fouling resistance at time t = 1 day, with mean $\mu(R_f(1))$ and standard deviation $\sigma(R_f(1))$.
- β : is assumed constant for this model, and all randomness in $R_f(t)$ is due to randomness in $R_f(1)$.

In order to find the average value of the fouling growth $R_f(t)$, the average value of $R_f(1)$ needs to be determined from the thermal analyses. Then, the only unknown in equation (5.10) will be the value of β, which could be determined using the condition that the critical level of fouling will correspond to the mean time to reach failure of the heat exchanger (i.e., performance failure due to fouling). The critical level of fouling is determined from the thermal analyses (i.e., representing average of endpoint fouling for several fouling cycles), plant experience, or international standards.

Solution Procedure:

The flow chart of the developed program for the statistical analysis method is shown in Figure 5.1. The input data of the statistical analysis program included: the end point failure for time to clean due to fouling for many fouling cycles, i.e., the critical level of fouling and the fouling level after one day of operation. The output of the program is the statistical fouling model

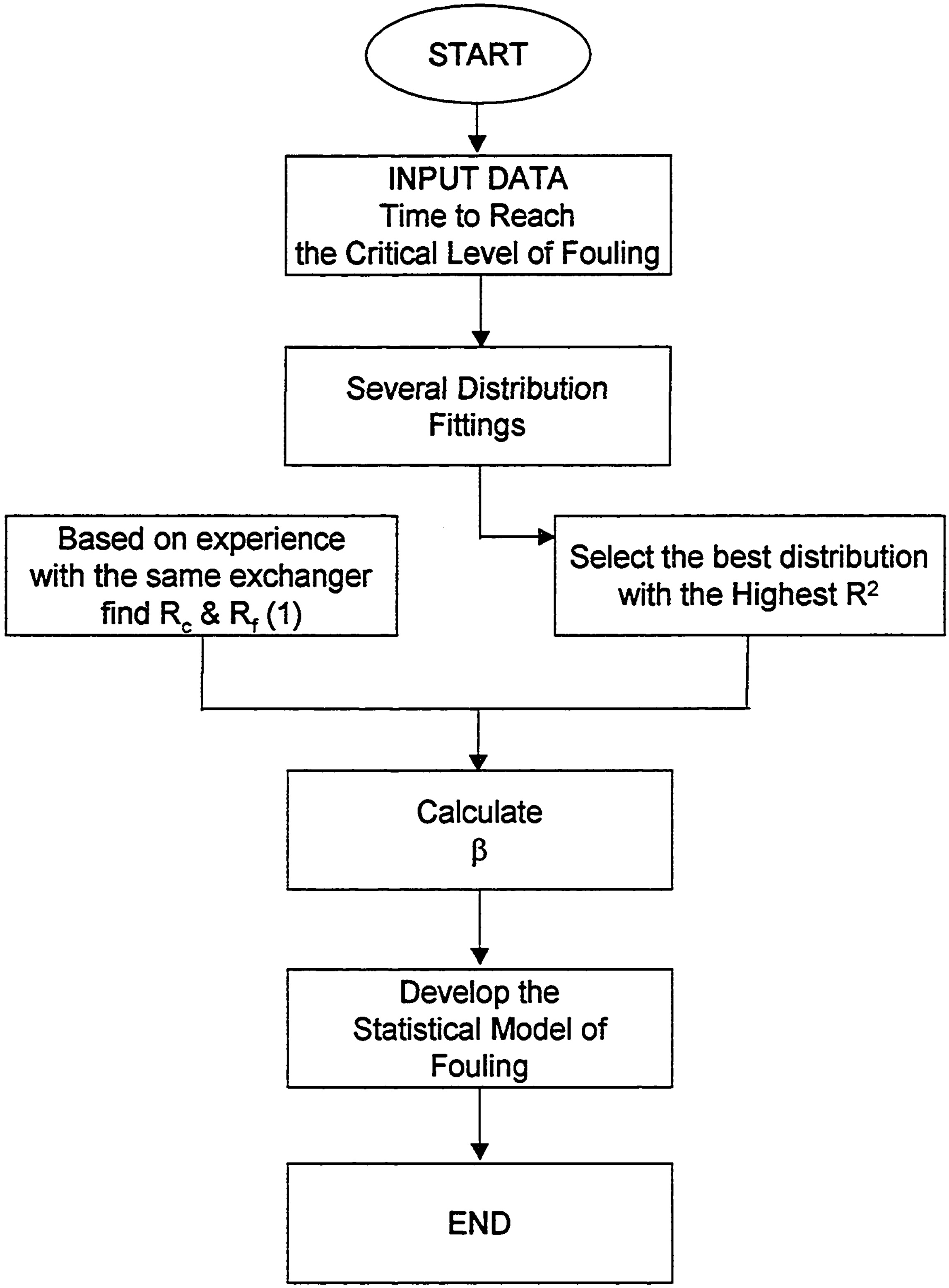


Figure 5.1: Flow Chart for Statistical Evaluation of Fouling

that could be used to predict the fouling growth in the exchanger under consideration.

5.2 Uncertainty Analysis

Using the statistical analysis, the fouling resistance when time to clean is lognormally distributed is given as:

$$R_f(t)=R_f(1)\,t^{\beta} \tag{5.11}$$

where: $R_f(1)$ *and* β are constant for a given exchanger. $R_f(1)$ is the average value after one day of operation.

Using equation (4.81), the error in R_f is given as:

$$\Delta^2 R_f(t) = \left[R_f(1) \quad \beta\, t^{(\beta-1)}\right]^2 \Delta^2 t \tag{5.12}$$

where Δt is calculated form the confidence interval of the given distribution.

For exchanger case 1 and using the above procedure and Δt value from chapter 4, it can easily be shown that the error in R_f using the statistical analysis method will be about ±0.2%.

5.3 Results and Discussion

The fouling prediction results using the statistical approach presented in this study for both exchangers case 1 and case 2 are given in the following sections:

5.3.1 Exchanger Case 1

This set of results pertains to the prediction of the critical level of fouling using statistical approach described earlier in section 5.1. Such approach, which is used by researchers [6, 36, 37, 92], is a good tool to determine the actual time required to reach to the critical level of fouling. The fouling cycles data for exchanger case 1 has been analyzed statistically and the results are shown in Tables A5 and A6. Table A5 shows the statistical functions and Table A6 shows the results for different distribution functions that could represent the fouling failure of the subject exchanger. The same statistical functions shown in Table A6 are represented graphically in Figure 5.2. As can be seen, the reliability of the subject exchanger is decreasing with time. The distribution data shown in Table A6 is plotted in Figures 5.3a and 5.3b and each data has been fitted with straight line. The fitted equation, the corresponding coefficient of determination, and the goodness for fit test results (Kolmogorov-Smirnov (K-S) test [93]) are given in Table 5.2. The K-S tests indicate that Weibull, Normal and Lognormal distributions have good fit for the data since their K-S modified test statistics are less than the modified critical values. This conclusion supports the judgment based on the coefficient of determination. Since the coefficient of determination is the highest for the lognormal distribution (R^2 = 0.9175), it has been selected to develop the statistical fouling growth model. The 95% confidence interval of R^2 for the selected lognormal distribution is the highest (0.6068 to 0.9852).

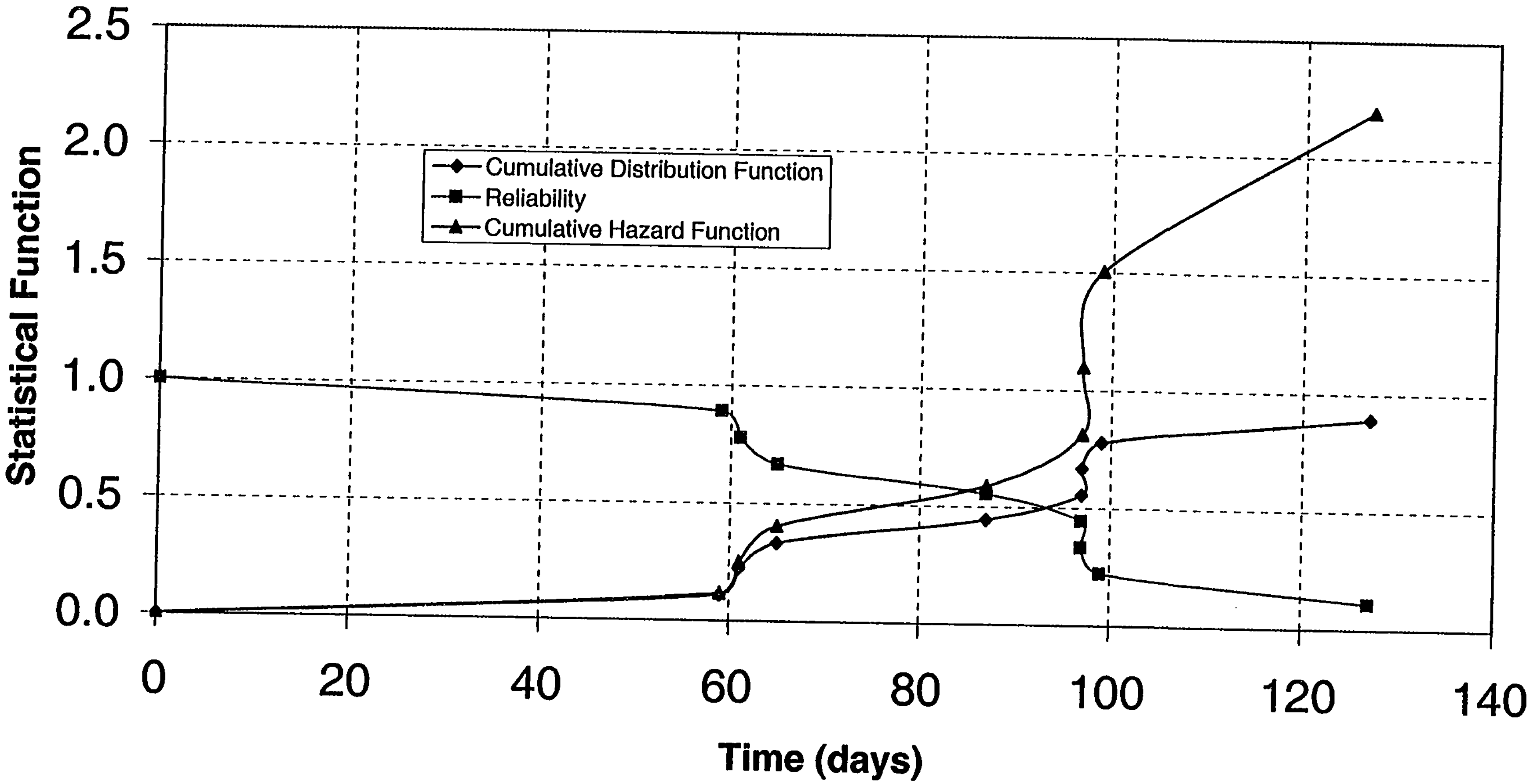


Figure 5.2: Statistical Functions of the Fouling for the Heat Exchanger Case 1

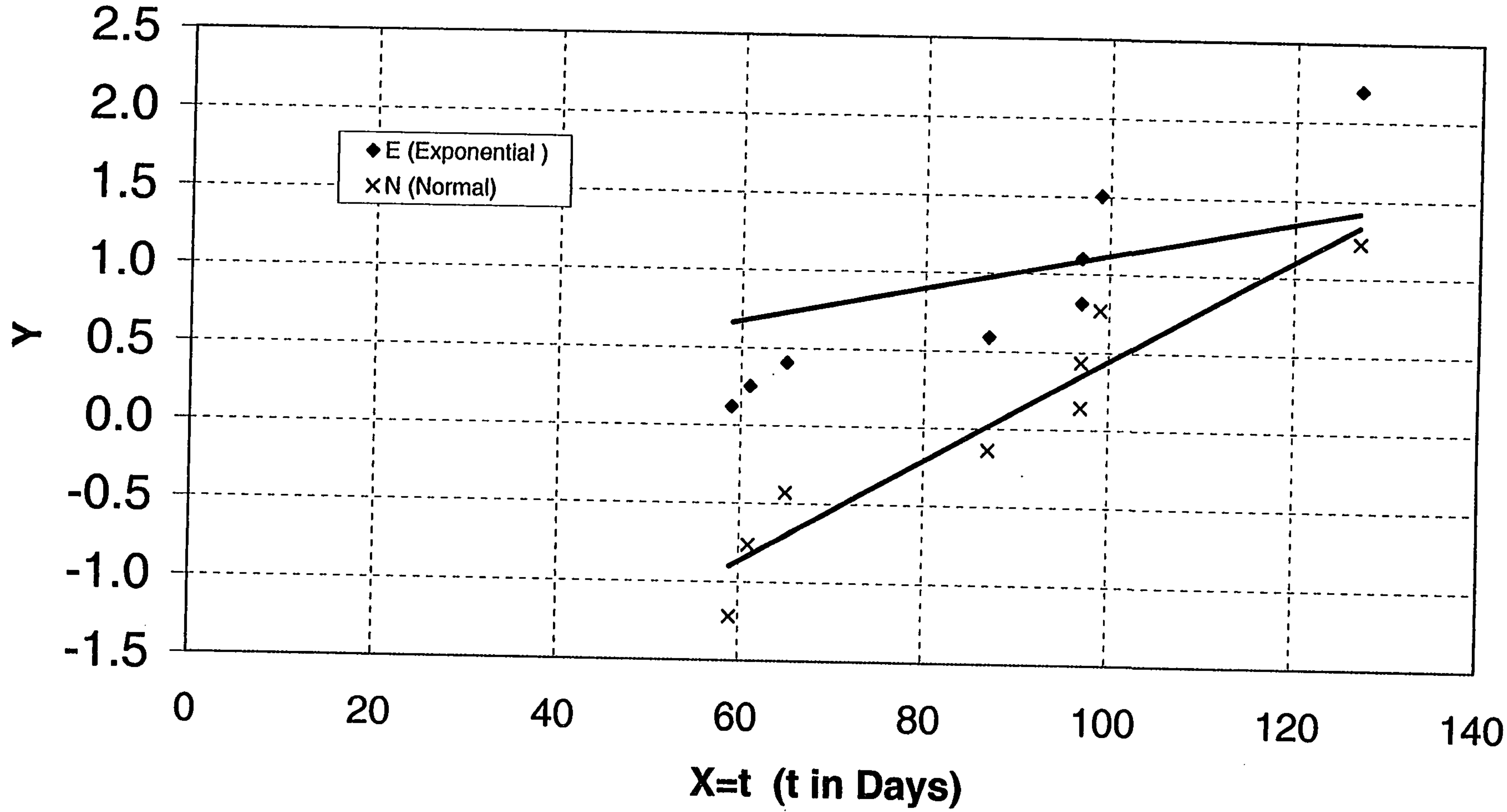


Figure 5.3a : Exponential and Normal Distributions for Fouling in Heat Exchanger Case 1

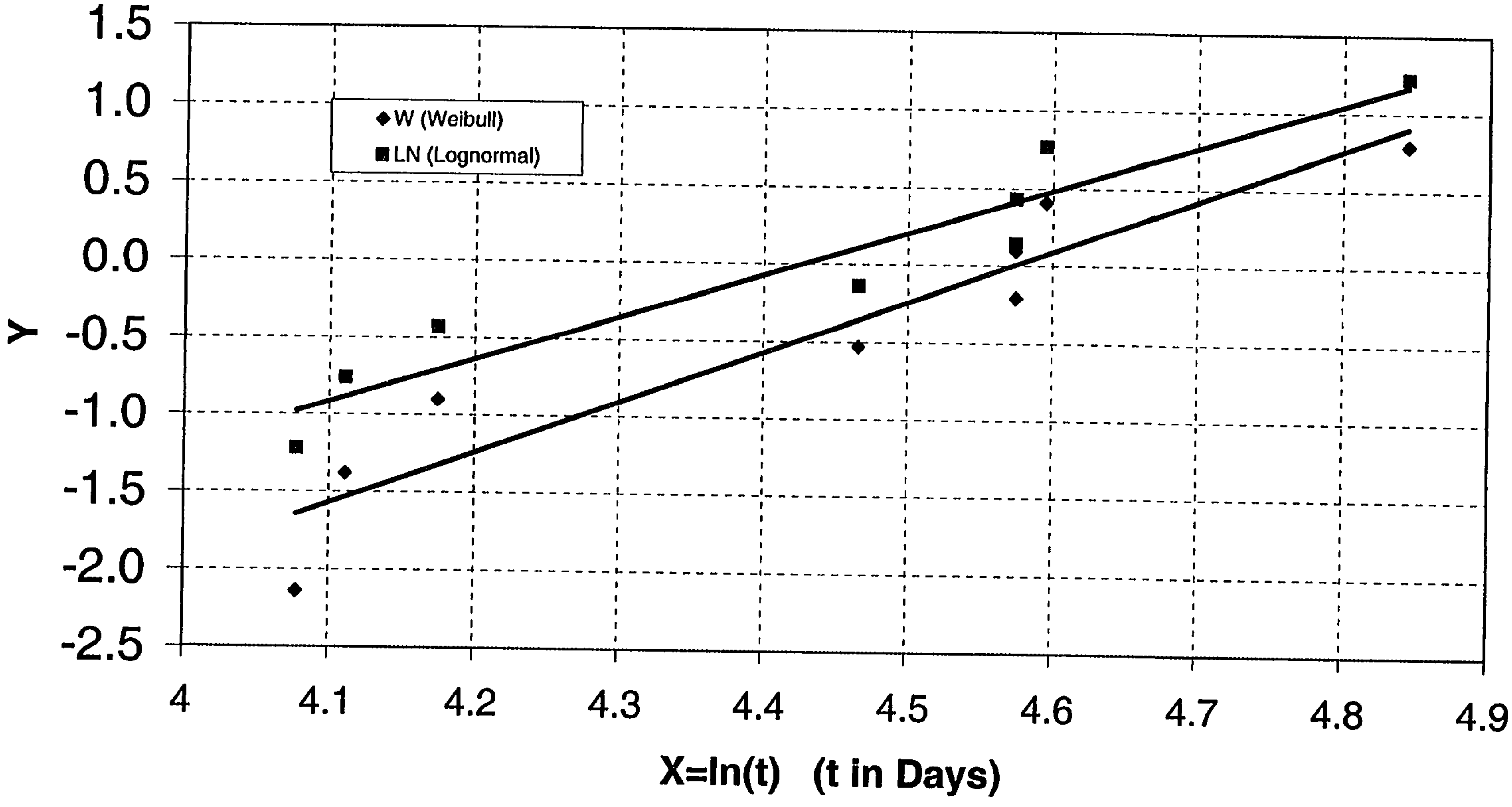


Figure 5.3b : Weibull and Lognormal Distributions for Fouling in Heat Exchanger Case 1

Table 5.2: Summary of the Statistical Distribution Straight Line Fit Results (Heat Exchanger Case 1)

Distribution	**Fitted Equation**	**Coefficient of Determination (R^2)**	**Goodness of Fit Test**	
			K-S Modified Test Statistic	**Modified Critical Value**
Exponential (E)	Y= 0.0112 X	0.5474	1.053	0.819
Weibull (W)	Y= 3.3227 X–15.194	0.9000	0.335	0.819
Normal (N)	Y= 0.0327 X–2.8324	0.9138	0.376	.0895
Lognormal (LN)	Y= 2.8008 X–12.4	0.9175	0.326	0.895

The average value of the fouling growth could is determined using the procedure described earlier in section 5.1 and the following data:

- The average value of the fouling $R_f(1)$ for the subject exchanger case 1 is found using the results of the thermal analyses for the fouling cycles. This value is found to be 0.002829 hr.ft^2.F/Btu (0.000498 m^2 C/W).
- The mean time to reach to the critical level of fouling (μ_c) is calculated for the best distribution (lognormal distribution for the exchanger under consideration) from equation (5.9). This value is found to be 89.2 days. Based on the thermal analyses and the experience in similar services, the critical level of fouling is 0.015 hr.ft^2.F/Btu (0.00264 m^2 C/W). This critical level of fouling for this particular exchanger will be reached at an average cleaning time of 89.2 days.

Based on the above data and the fact the lognormal distribution can be derived from a power law function $R_f(t) = R_f(1)\ t^\beta$ which characterizes the random fouling growth process [36], the constant β is found to be 0.3715. Thus the average fouling growth for exchanger case 1 using the statistical approach presented in this study will be:

$$R_f(t) = 0.002829\ t^{0.3715}$$

To compare the results, all thermal analyses results for the 8 fouling cycles are superimposed on each other in Figure 5.4 and then fitted by a single representative power law equation using regression. The average fouling growth using these thermal analysis results is found to be:

$$R_f(t) = 0.0031\ t^{0.369}$$

The comparison between the average fouling growth results using the statistical approach presented in this study and regression fitting for all thermal results (this requires the detail thermal analyses with respect to time be done before this regression fitting) is shown in Table 5.3. From the table, it is clear that the maximum difference in the average fouling growth is between the two approaches is about 8.7%. Therefore, the statistical approach represents an easier approach to predict fouling growth using thermal analysis only at the end points as compared to the detailed thermal analyses done at all points in time.

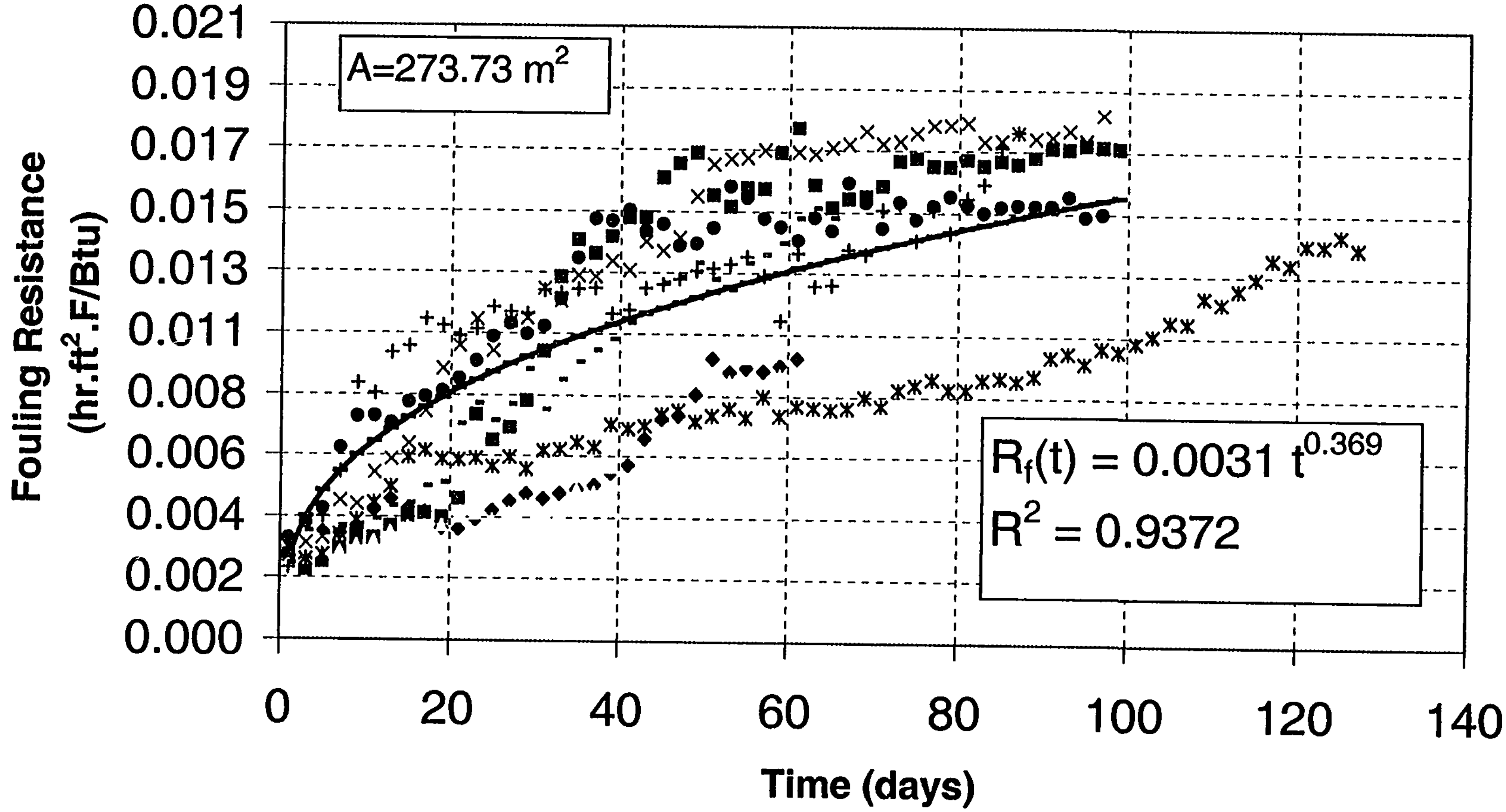


Figure 5.4: Fouling Growth Results for All fouling Cycles for Exchanger Case 1

Table 5.3: Comparison of the Statistical Models for Exchanger Case 1

Time (days)	$R_f(t)$ from the statistical approach presented in this study	$R_f(t)$ from the regression fitting after the performance based on detail thermal analyses for all fouling cycles	% Difference
1	0.00283	0.00310	8.7
5	0.00514	0.00561	8.4
10	0.00665	0.00725	8.2
15	0.00774	0.00842	8.1
30	0.01001	0.01087	8.0
40	0.01114	0.01209	7.9
50	0.01210	0.01313	7.8
60	0.01295	0.01404	7.8
70	0.01371	0.01487	7.8
80	0.01441	0.01562	7.7
90	0.01505	0.01631	7.7

The fouling growth predicted using the developed statistical model is compared with those obtained using thermal analysis. The comparison is shown graphically in Figure 5.5 for three fouling cycles. The t and F tests results for the thermal analyses (BD) and statistical model results are given in Table 5.4. It could be concluded that they give same results and similar variability.

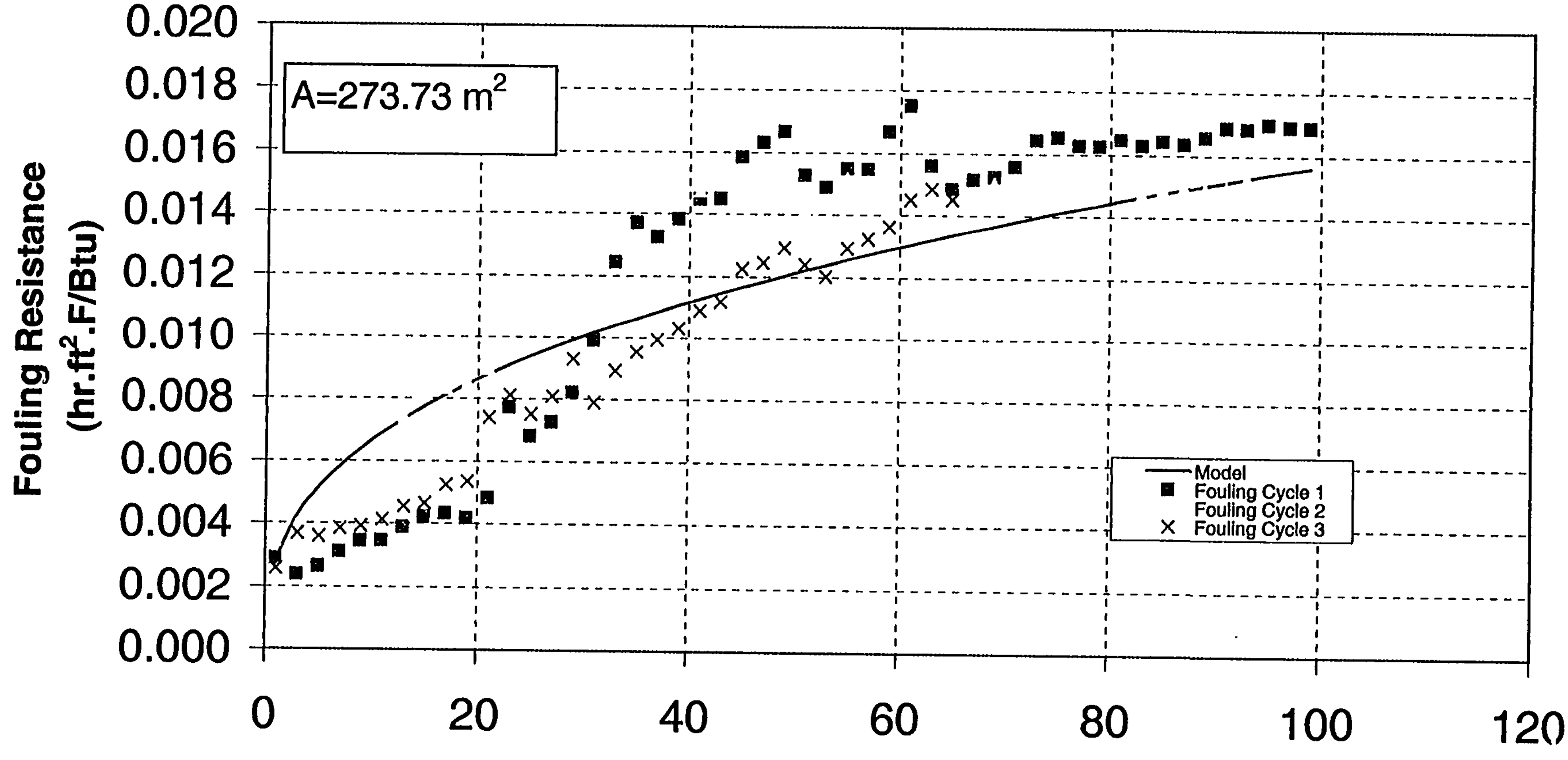


Figure 5.5: Comparison Between Fouling Growth Results for Exchanger Case 1

Table 5.4: Comparison Between Fouling results Obtained from Thermal and Statistical Model for Exchanger Case 1

	Fouling Cycle 1		**Fouling Cycle 2**		**Fouling Cycle 3**	
	Thermal	Model	Thermal	Model	Thermal	Model
Mean	0.0123	0.0116	0.0123	0.00116	0.00897	0.00997
Variance	2.8369×10^{-5}	1.11229×10^{-5}	1.175×10^{-5}	1.095×10^{-5}	1.4598×10^{-6}	8.171×10^{-6}
Number of sample points (n)	50	50	49	49	33	33
Common standard deviation (S_p)	0.0044		0.00337		0.00337	
Test statistic (t_o)	0.696		1.134		1.201	
Test statistic (F_o)	2.550		1.072		1.786	
$t_{\alpha/2,n_1+n_2-2}$	1.987		1.988		2.042	
$F_{\alpha/2,n_1-1,n_2-1}$	1.880		1.880		2.070	
Conclusion	All cycles indicated that the fouling results are same based on the thermal and statistical models. Also the variability is the same (with exception to fouling cycle 1).					

It could be concluded that the statistical method is a good tool for predicting the time required to reach critical fouling level and easier than performing detailed thermal analysis. However, there is a need to know the critical fouling value from previous experience for the exchanger under consideration (exchanger Case 1).

5.3.2 Exchanger Case 2

This set of results pertains to the prediction of the critical level of fouling for exchanger case 2 using statistical approach. The fouling cycles data has been analyzed statistically and the results are shown in Tables B4 and B5. Table B4 shows the calculated statistical functions for the fouling cycles of the subject heat exchanger case 2. The same statistical functions shown in Table B4 are represented graphically in Figure 5.6. As can be seen, the reliability of the subject exchanger is decreasing with time. Table B5 shows calculated different distribution functions that could represent the fouling failure of the subject exchanger. The distribution data shown in Table B5 is plotted in Figures 5.7a and 5.7b and each data has been fitted with straight line. The fitted equation, the corresponding coefficient of determination, and the goodness for fit test results (K-S test [93]) are given in Table 5.5. The K-S tests indicate that Weibull, Normal and Lognormal distributions have good fit for the data since their K-S modified test statistics are less than the modified critical values. This conclusion supports the judgment based on the coefficient of determination. Since the coefficient of determination is the highest for the lognormal distribution (R^2 = 0.8947), it has been selected to develop the statistical fouling growth model. The 95% confidence interval of R^2 for the selected lognormal distribution is the highest (0.6398 to 0.9726).

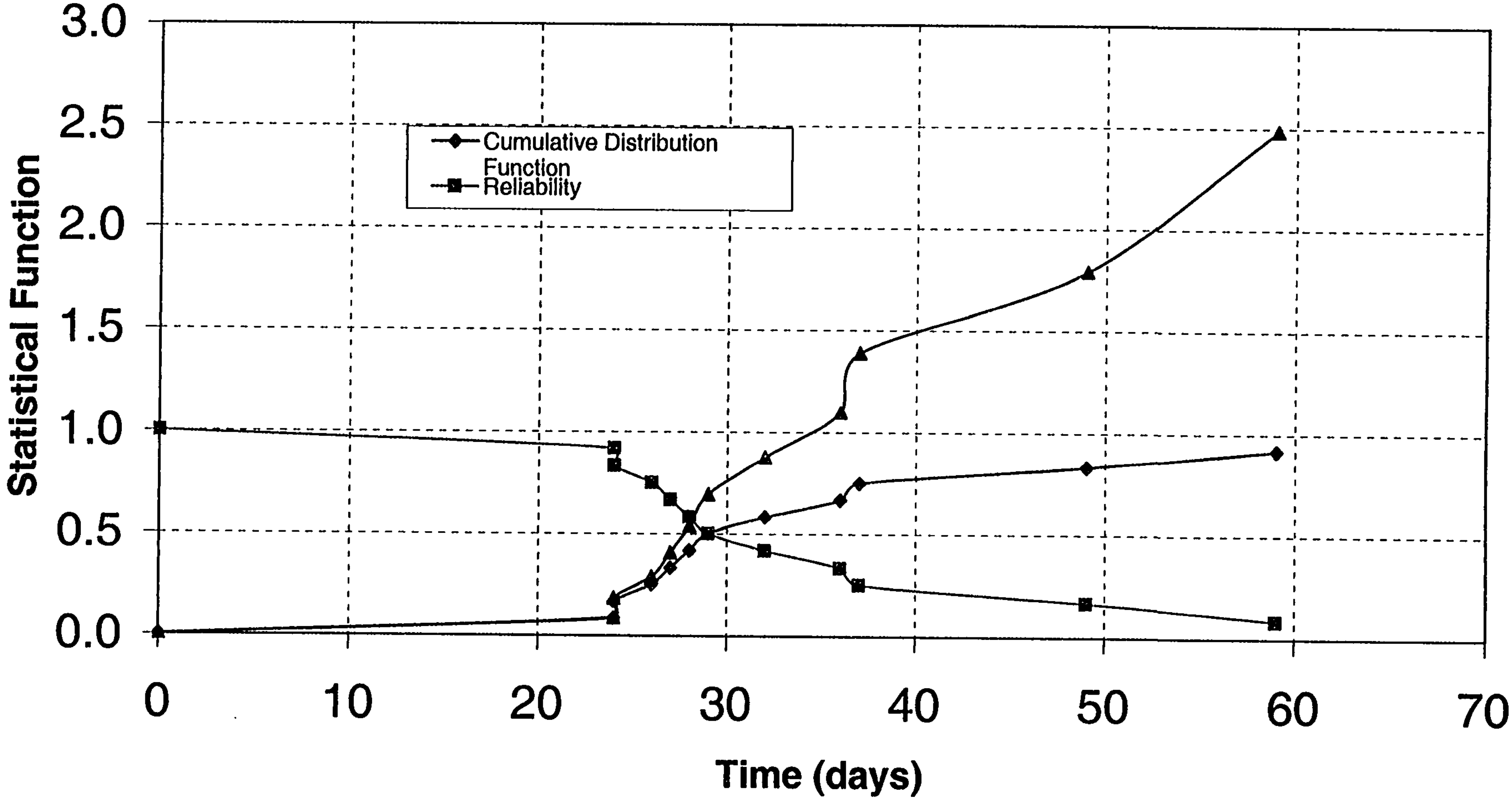


Figure 5.6: Statistical Functions of the Fouling for the Heat Exchanger Case 2

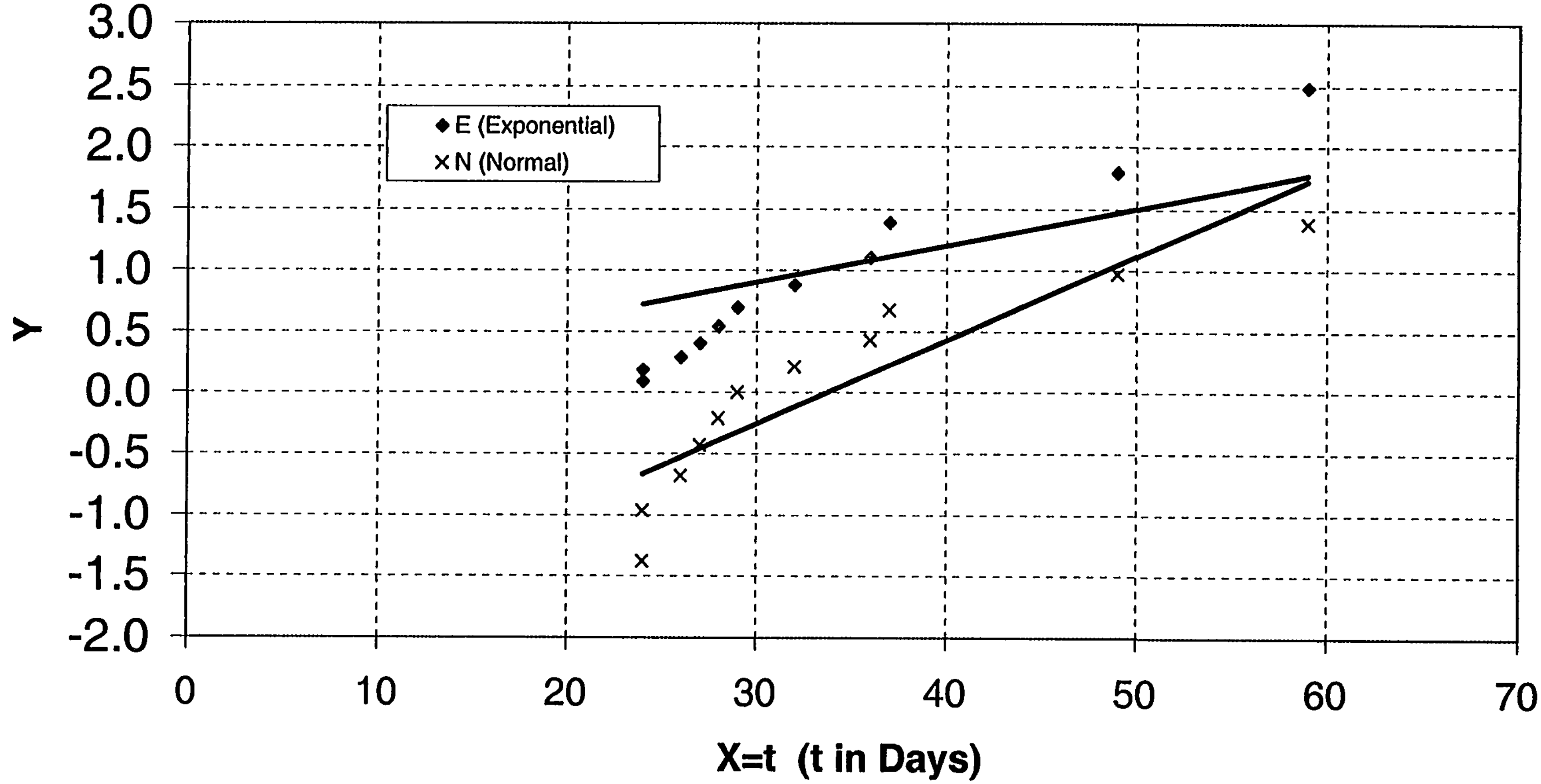


Figure 5.7a: Exponential and Normal Distributions for Fouling in Heat Exchanger Case 2

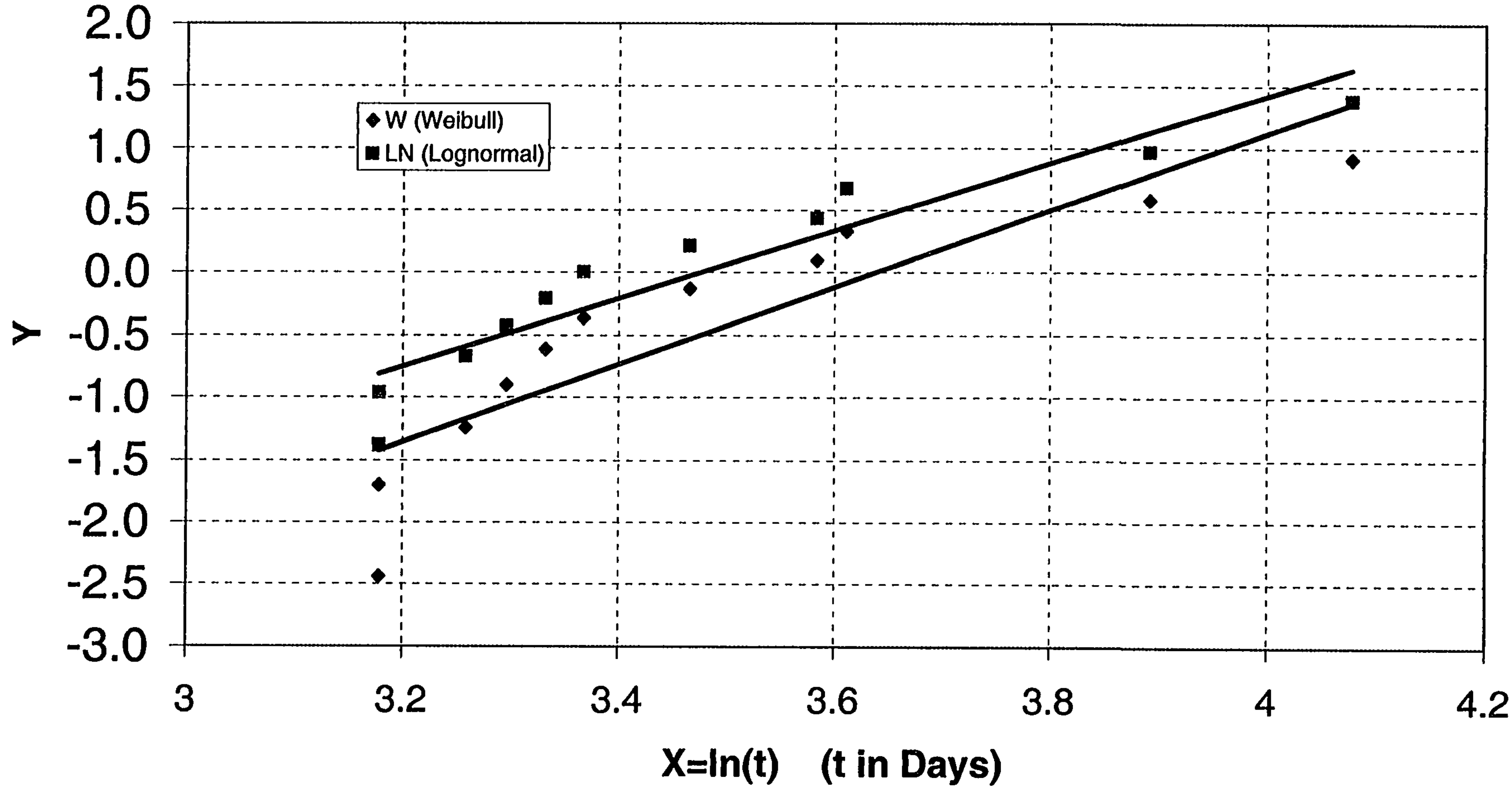


Figure 5.7b: Weibull and Lognormal Distributions for Fouling in Heat Exchanger Case 2

Table 5.5: Summary of the Statistical Distribution Straight Line Fit Results (Heat Exchanger Case 2)

Distribution	**Fitted Equation**	**Coefficient of Determination (R^2)**	**Goodness of Fit Test**	
			K-S Modified Test Statistic	**Modified Critical Value**
Exponential (E)	Y= 0.0301 X	0.6549	1.465	0.819
Weibull (W)	Y= 3.0945 X–11.257	0.7947	0.493	0.819
Normal (N)	Y= 0.0683 X–2.3042	0.8191	0.604	.0895
Lognormal (LN)	Y= 2.72 X–9.4554	0.8947	0.446	0.895

The average value of the fouling growth is calculated using the procedure described in section 5.1 and the following information:

- The average value of the fouling $R_f(1)$ for the subject exchanger case 2 is found using the results of the thermal analyses for the fouling cycles. This value is found to be 0.001 hr.ft^2.F/Btu (0.000176 m^2 C/W).
- The mean time to reach to the critical level of fouling (μ_c) is calculated for the best distribution (lognormal distribution for the exchanger under consideration) from equation (5.9). This value is found to be 34.6 days. Based on the thermal analyses and the experience in similar services, the critical level of fouling is 0.0038 hr.ft^2.F/Btu (0.0006688 m^2 C/W). This critical level of fouling for this particular exchanger will be reached at average cleaning time of 34.6 days.

Based on the above data and the fact the lognormal distribution can be derived from a power law function $R_f(t) = R_f(1)\, t^{\beta}$ which characterizes the random fouling growth process [36], the constant β is found to be 0.3767. Thus the average fouling growth for exchanger case 2 using the statistical approach presented in this study will be:

$$R_f(t) = 0.001\ t^{0.3767}$$

To compare the results, the thermal analyses results for the 11 fouling cycles are superimposed on each other in Figure 5.8 and then fitted by a single representative power law equation using regression. The average fouling growth using these thermal analyses results is found to be:

$$R_f(t) = 0.00095\ t^{0.3942}$$

The comparison between the two approaches is shown in Table 5.6 with a maximum difference of 6%. This indicates developed statistical approach is very effective tool to monitor the heat exchanger fouling growth instead of details thermal analyses.

Table 5.6: Comparison of the Statistical Models for Exchanger Case 2

Time (days)	$R_f(t)$ from the statistical approach presented in this study (hr.ft^2.F/Btu)	$R_f(t)$ from the regression fitting after the performance based on detail thermal analyses for all fouling cycles (hr.ft^2.F/Btu)	% Difference
1	0.0010	0.0009	6.0
5	0.0018	0.0018	3.3
10	0.0024	0.0023	2.1
20	0.0031	0.0031	0.9
30	0.0036	0.0036	0.2
35	0.0038	0.0038	0.1

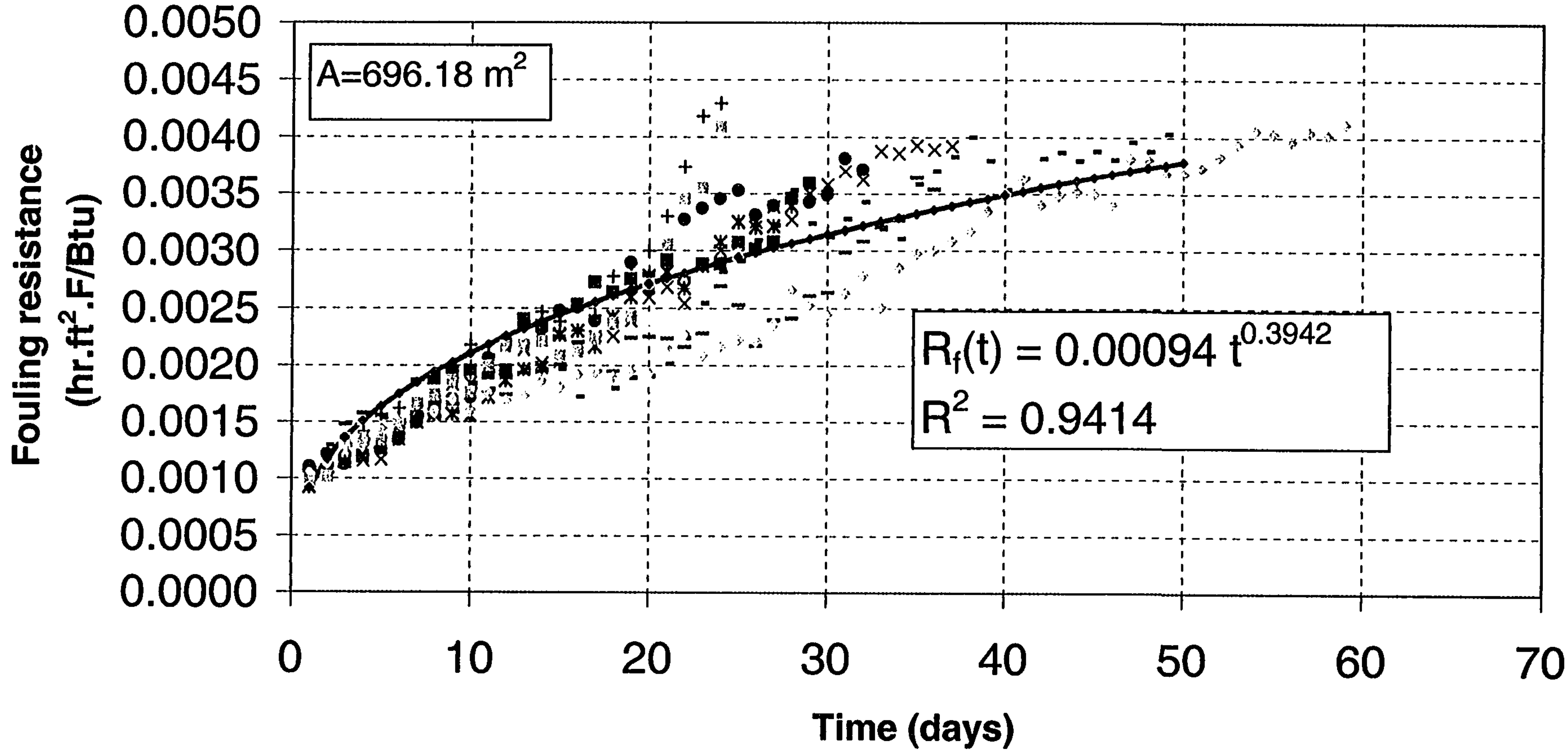


Figure 5.8: Fouling Growth Results for All Fouling Cycles for Exchanger Case 2

The fouling growth predicted using the developed statistical model is compared with those obtained using thermal analysis. The comparison is shown graphically in Figure 5.9 for two fouling cycles. The t and F tests results for the thermal analyses and statistical model results are given in Table 5.7. It could be concluded that they give same results and similar variability.

Table 5.7: Comparison Between Fouling results Obtained from Thermal and Statistical Model for Exchanger Case 2

	Fouling Cycle 1		**Fouling Cycle 2**	
	Thermal	Model	Thermal	Model
Mean	0.002282	0.002633	0.0025295	0.002875
Variance	5.56497×10^{-7}	4.96532×10^{-7}	8.7432×10^{-7}	6.0528×10^{-7}
Number of sample points (n)	29	29	37	37
Common stand- ard deviation (S_p)	0.00072		0.00086	
Test statistic (t_o)	1.847		1.732	
Test statistic (F_o)	1.121		1.444	
$t_{\alpha/2,n_1+n_2-2}$	2.004		1.996	
$F_{\alpha/2,n_1-1,n_2-1}$	2.110		1.890	
Conclusion	The comparison indicate that the two methods give same fouling results and they have similar variability.			

As concluded in exchanger case 1 and just shown for exchanger case 2, it could be seen that the statistical method is a good tool for predicting the time required to reach critical fouling level and easier than performing detailed thermal analysis. However, there is a need to know the critical fouling value from previous experience for the exchanger under consideration.

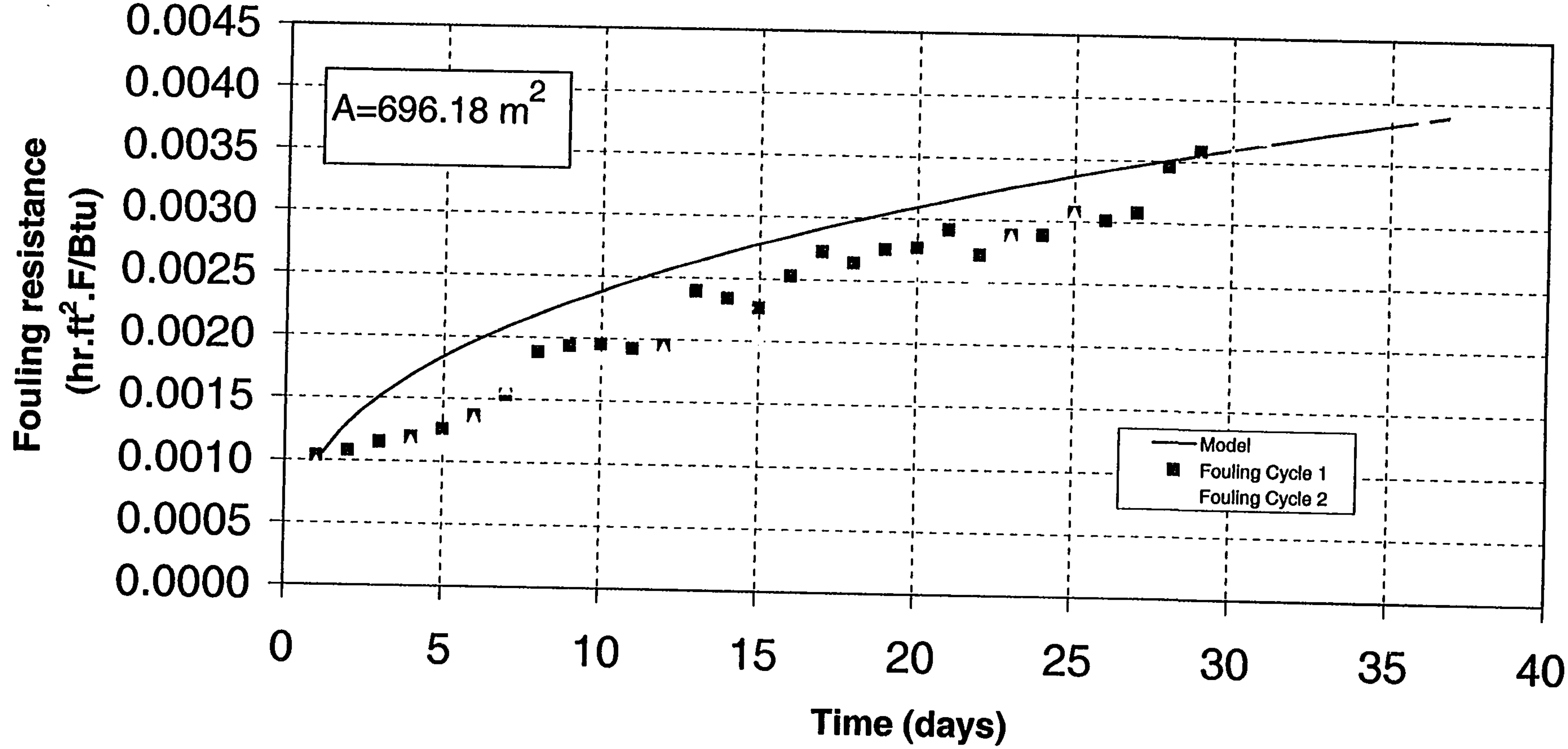


Figure 5.9: Comparison Between Fouling Growth Results for Exchanger Case 2

CHAPTER 6

FOULING AND IRREVERSIBILITY

In this chapter, using the second law of thermodynamics and heat exchanger thermal analysis, a relationship between the irreversibly and fouling resistance is developed. As an example, the effect of fouling on the irreversibility for exchanger case 2 is presented and discussed in this chapter.

6.1 Fouling and Irreversibility Relationship

In this section, the second law of thermodynamics will be used to develop a relationship between irreversibility and fouling resistance. Such relationship gives an indication of the level of fouling in a heat exchanger. It has been reported in the literature [47] that an increase of irreversibility indicates an increase of fouling level at the same thermal and hydraulic conditions. The advantage of this method is that it is very easy as compared to the other methods and could be used easily by the plant's engineer to monitor the fouling level without going into detail analysis.

To examine the irreversibility effects on fouling in heat exchangers, a relationship between these two parameters has been developed. The first step in developing such a relationship is to develop an expression for irreversibility in terms of both fluid streams exit temperatures.

The second law of thermodynamics for a control volume is given by [94]:

$$\frac{dS_{c.v}}{dt}+\sum_{out}\dot{m}s-\sum_{in}\dot{m}s-\sum_{in}\frac{\dot{Q}}{T}=\dot{S}_{gen}\geq 0 \tag{6.1}$$

where:

s	:	the entropy, kJ/kg K
$\dot{Q}$	:	The total rate of heat transfer from or to the control volume, W
$\dot{m}$	:	mass flow rate , kg/s
$\dot{S}_{gen}$	:	rate of entropy generation, W/K

Assuming a heat exchanger undergoing a steady state steady flow process with heat transfer taking place between the two fluid streams only (i.e. $\dot{Q}=0$), it can be shown using equation (6.1) that:

$$\dot{S}_{gen}=(\dot{m}s)_{out}-(\dot{m}s)_{in}$$

or

$$\begin{aligned}\dot{S}_{gen}&=\dot{m}_t\left(s_{t,out}-s_{t,in}\right)+\dot{m}_s\left(s_{s,out}-s_{s,in}\right)\\ \dot{S}_{gen}&=\dot{m}_t\Delta s_t+\dot{m}\Delta s_s\end{aligned} \tag{6.2}$$

Using the T-ds relations and assuming no phase change and negligible pressure drop along the fluid stream, the entropy generation can be written as:

$$\dot{S}_{gen}=\dot{m}_t C_{p,t}\,\ell n\left(\frac{T_{t,out}}{T_{t,in}}\right)+\dot{m}_s C_{p,s}\,\ell n\left(\frac{T_{s,out}}{T_{s,in}}\right) \tag{6.3}$$

Since the rate of irreversibility change I is given by :

$$I=T_o\dot{S}_{gen} \tag{6.4}$$

Hence, using equation (6.3), the irreversibility is written in terms of fluid streams exit temperatures as:

$$I = T_o\left[\dot{m}_t C_{p,t}\,\ell n\left(\frac{T_{t,out}}{T_{t,in}}\right) + \dot{m}_s C_{p,s}\,\ell n\left(\frac{T_{s,out}}{T_{s,in}}\right)\right] \tag{6.5}$$

For two phase heat exchangers

$$\Delta s = \frac{h_{lv}}{T_{cond}} \quad \textit{for condensation}, J/kg\,K$$

$$\Delta s = \frac{h_{lv}}{T_{evap}} \quad \textit{for evaporation}, J/kg\,K$$

where:

h_{lv} : latent heat of evaporation, J/kg

T_c : condensation temperature, C

T_{evap} : evaporation temperature, C

Equation (6.5) indicates that the rate of irreversibility is a function of heat exchangers exit temperatures. To relate I to the fouling resistance, a relation between the fouling resistance and the heat exchangers exit temperatures need to be derived. For a given exchanger, the inlet temperatures are given and fixed. The outlet temperatures will vary based on the exchanger fouling conditions. With the increase of fouling, the heat transfer rate decreases and consequently the outlet temperatures will be affected.

Performing an energy balance along the fluid streams, the outlet temperatures can be rewritten as:

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{(\dot{m}C_p)_h} \tag{6.6}$$

$$T_{c,out}=T_{c,in}+\frac{\dot{Q}}{\left(\dot{m}C_p\right)_c} \tag{6.7}$$

Where the heat transfer rate ($\dot{Q}$) is given in terms of the thermal effectiveness (ε) as:

$$Q=\varepsilon Q_{\max}=\varepsilon\left(\dot{m}C_p\right)_{\min}\left(T_{h,in}-T_{c,in}\right) \tag{6.8}$$

For 1-2 heat exchanger, the effectiveness is given by [25]:

$$\varepsilon=\frac{2}{1+c^{*}+\sqrt{1+c^{*2}}\left(1+e^{-\Gamma}\right)/\left(1-e^{-\Gamma}\right)} \tag{6.9}$$

where:

$$\Gamma=NTU\sqrt{1+C^{*2}} \tag{6.10}$$

$$C^{*}=\frac{\left(\dot{m}C_p\right)_{\min}}{\left(\dot{m}C_p\right)_{\max}} \tag{6.11}$$

And the minimum number of transfer units (NTU) is given as:

$$NTU_{\min}=\frac{AU}{\left(\dot{m}C_p\right)_{\min}} \tag{6.12}$$

For two-phase heat exchangers (with negligible pressure drop), the effectiveness is given as follow:

$$\varepsilon=1-\exp\left(-NTU\right) \tag{6.13}$$

The fouling, resistance (R_f) is written in terms of the overall heat transfer coefficient U as:

$$U_a=\frac{1}{\frac{1}{h_s}+\frac{D_o}{2k}\ln\left(\frac{D_o}{D_i}\right)+\frac{D_o}{h_t D_i}+R_f(t)} \tag{6.14}$$

Substituting equations (6.8) to (6.14) into equations (6.6) and (6.7), the two fluid streams exit temperatures can be expressed in terms of the fouling resistance. Using these exit temperatures with equation (6.5), the rate of irreversibility change can be expressed in terms of the fouling resistance.

Solution Procedure:

The flow chart of the developed program for the second law analysis of heat exchanger subject to fouling is shown in Figure 6.1

6.2 Uncertainty Analysis

The input data for the second-law analysis program included: inlet and outlet temperatures, flow rates and properties for both fluids. Samples of these data are shown in Table 6.1 and Table B2.

Table 6.1: Samples of the Collected Data for Exchanger Case 2

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	10/15/99	156.14	74.62	55.18	75.07	12.18	745.30
2	10/16/99	157.59	75.99	55.72	74.67	12.73	817.91
3	10/17/99	157.21	78.16	55.41	74.79	12.47	784.17
4	10/18/99	157.25	78.82	56.54	75.72	12.87	815.74
5	10/19/99	157.33	77.97	58.04	75.49	12.84	892.37
6	10/20/99	157.36	77.41	58.15	75.80	12.84	882.30

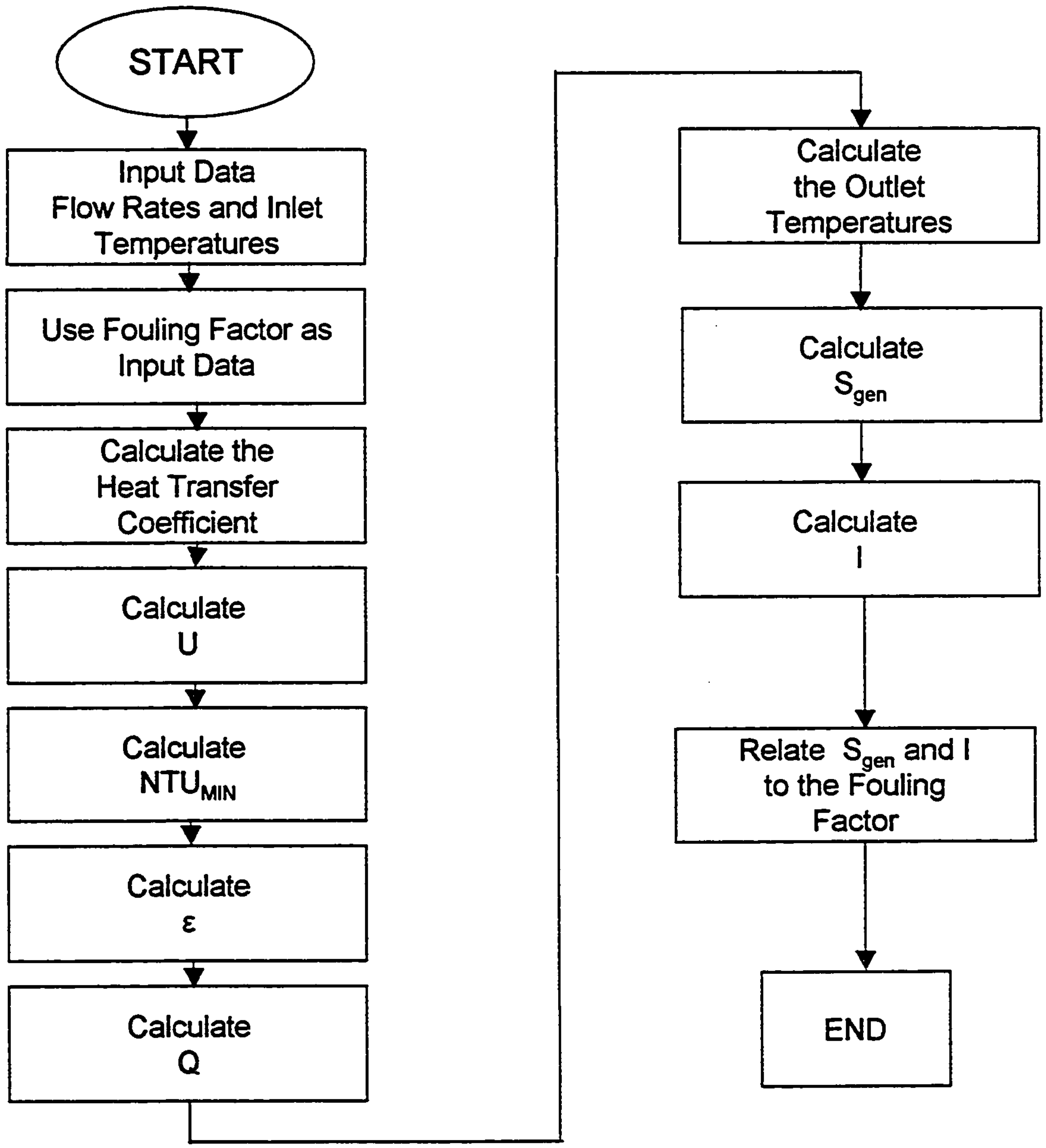


Figure 6.1: Flow Chart for Heat Exchanger Fouling Evaluation Based on the Second Law of Thermodynamics

The program calculates both irreversibility and entropy generation for both flow streams. Moreover, the program calculates the total entropy generation and irreversibility and relates them to the fouling growth.

The entropy generation, $\dot{S}_{gen}$, is given as:

$$\dot{S}_{gen} = \dot{m}_t C_{p,t} \ell n\left(\frac{T_{t,out}}{T_{t,in}}\right) + \dot{m}_s C_{p,s} \ell n\left(\frac{T_{s,out}}{T_{s,in}}\right) \tag{6.15}$$

Using equation (4.81), the error in $\dot{S}_{gen}$ is given as:

$$\begin{aligned}\Delta\dot{S}_{gen}^2 &= \left(\frac{\partial \dot{S}_{gen}}{\partial \dot{m}_t}\right)^2 \Delta\dot{m}_t + \left(\frac{\partial \dot{S}_{gen}}{\partial \dot{m}_s}\right)^2 \Delta\dot{m}_s + \left(\frac{\partial \dot{S}_{gen}}{\partial T_{t,out}}\right)^2 \Delta T_{t,out} \\ &+ \left(\frac{\partial \dot{S}_{gen}}{\partial T_{t,in}}\right)^2 \Delta T_{t,in} + \left(\frac{\partial \dot{S}_{gen}}{\partial T_{s,out}}\right)^2 \Delta T_{s,out} + \left(\frac{\partial \dot{S}_{gen}}{\partial T_{s,in}}\right)^2 \Delta T_{s,in}\end{aligned} \tag{6.16}$$

Consider the data given in Table B2 and the instruments' inaccuracies in chapter 4:

$\Delta\dot{m}_t = \Delta\dot{m}_s = 1.0\%$

$\Delta T_{t,out} = \Delta T_{t,in} = \Delta T_{s,out} = \Delta T_{s,in} = \Delta = 1.5\%$

If we substitute in equation (6.16), the error in $\dot{S}_{gen}$ will be given as:

$$\begin{aligned}\Delta^2 \dot{S}_{gen} = &\left\{\left[C_{p,t} \ell n\left(\frac{T_{t,out}}{T_{t,in}}\right)\right]^2 + \left[C_{p,s} \ell n\left(\frac{T_{s,out}}{T_{s,in}}\right)\right]^2 + \left[C_{p,t} \frac{T_{t,in}}{T_{t,out}}\right]^2 + \right. \\ &\left.\left[C_{p,t} \frac{T_{t,out}}{T_{t,in}}\right]^2 + \left[C_{p,s} \frac{T_{s,in}}{T_{s,out}}\right]^2 + \left[C_{p,s} \frac{T_{s,out}}{T_{s,in}}\right]^2\right\} \Delta^2\end{aligned} \tag{6.17}$$

It can be shown that the error in $\dot{S}_{gen}$ using the second law analysis will be about ±6.0%.

6.3 Results and Discussion

This set of results pertains to the use of irreversibility rate as an indication for fouling growth. As has been shown in the mathematical analysis section, there is a relation between the irreversibility rate and fouling growth. Fouling increases as irreversibly increases for heat exchangers with uniform inlet temperatures and mass flow rates for both fluid streams. The parameters required to calculate fouling growth using the second law analysis and the irreveriblity rate are flow rates and inlet and outlet tempertures for both streams.

Perforimng the second law analysis for exchanger case 2, samples of the results are shwon in Table 6.2 and Table B6. From the resulsts, it could be observed that irreversibility increases as fouling increases for heat exchangers with uniform inlet temperatures and mass flow rates for both fluid streams. This is shown in Figure 6.2, which shows the total irreversibility rate versus fouling growth. Similar findings are reported in the literature [47].

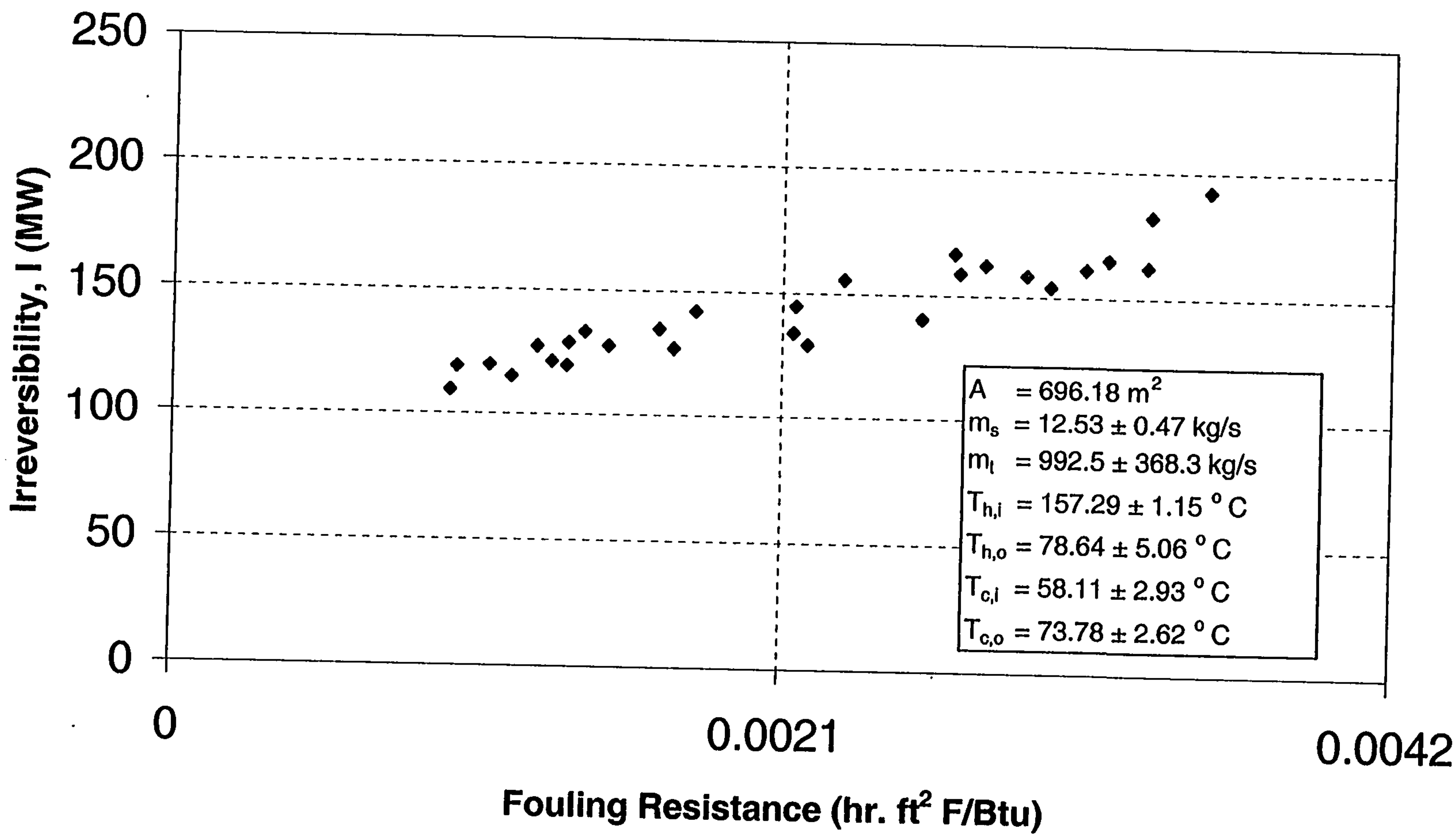


Figure 6.2: Total Irreversibility Versus Fouling for One Fouling Cycle for Exchanger Case 2

Table 6.2: Sample Results Using Second Law of Thermodynamics for Exchanger Case 2

Day	Date	$\dot{S}_{gen,tot}$ (W/ K)	I_{total} (W)	Fouling resistance (hr. ft2. F/BTU)
1	10/15/99	367603	1.10E+08	0.00096
2	10/16/99	401773	1.20E+08	0.00109
3	10/17/99	386407	1.15E+08	0.00117
4	10/18/99	399959	1.19E+08	0.00098
5	10/19/99	432827	1.29E+08	0.00136
6	10/20/99	427863	1.28E+08	0.00125

In order to relate the irreversibility to the critical level of fouling, the ratio of (I_{clean} / I), which is the irreversibility at the clean condition divided by the irreversibility, is plotted versus the fouling resistance as shown in Figure 6.3. From the figure, it is clear the irreversibility ratio (I_{clean}/I) decreases as the fouling increases. For exchanger case 2, the critical level of fouling is reached when the irreversibility ratio approach 0.55. Therefore, for each exchanger, the critical level of fouling could be linked to the irreversibility ratio. This finding could be used by the plants' engineers to continuously monitor the fouling growth in heat exchangers.

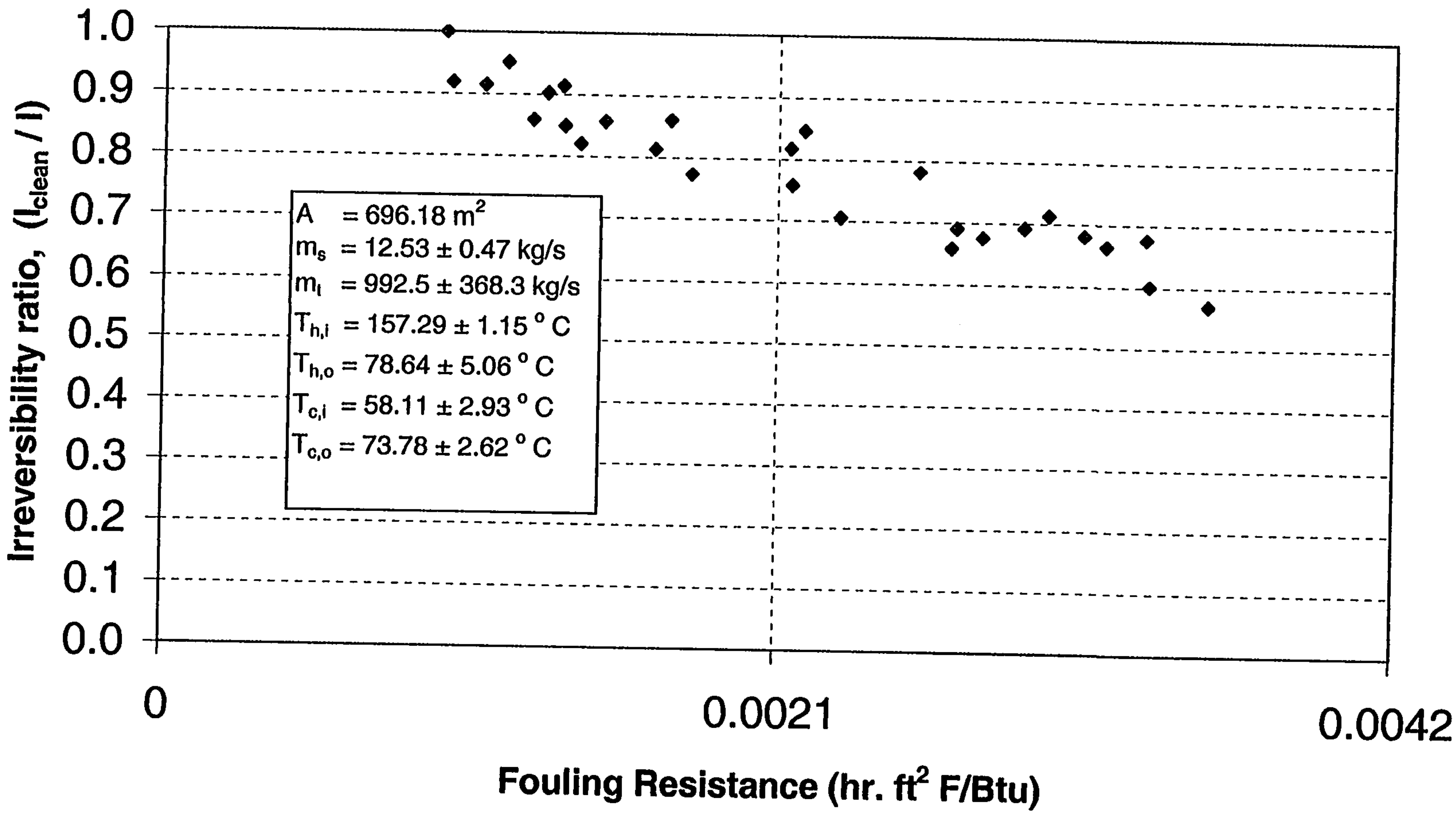


Figure 6.3: Irreversibility Ratio (I_{clean}/I) Versus Fouling for One Fouling Cycle for Exchanger Case 2

CHAPTER 7

FOULING MITIGATION

The literature review on fouling mitigation in heat exchangers indicates that a number of different methods are being used on the tube and shell sides. Recently new technologies in this regard have emerged, namely helical baffles for the shell-side and self-cleaning for the tube-side. It has been decided to introduce these fouling mitigation technologies in Saudi Aramco and to study their performance and effectiveness in mitigating fouling. This required the modification of two existing heat exchangers. The conventional baffle bundles in the naphtha heat exchanger have been replaced by helically baffled bundles. The thermosyphon reboiler has been modified and coupled with a self-cleaning test unit. The following sections contain a brief description for each mitigation technology, on line experiments and tested units, mathematical equations, and the results and discussion.

7.1 Mitigation Technologies

7.1.1 Helically Baffled Heat Exchanger

In order to solve the shell side-fouling problem in the naphtha heat exchanger (exchanger case 1), it has been decided to convert the existing naphtha heat exchangers to helically baffled heat exchangers and to study the effect of helical baffles on fouling mitigation.

Helical baffles as shown in Figure 7.1 are designed to guide the shell side flow into a helical pattern. Helical flow patterns do eliminate fouling and reduce the amount of the wasted energy by eliminating stagnation spots and eddies, and do reduce vibration by reducing unsupported tube spans. The helical flow pattern leaves no stagnant areas where fouling may accumulate. This new technology is considered a break through in the design of heat exchangers because it reduces maintenance and operation/energy cost and at the same time it reduces capital cost by reducing the heat exchanger's size (material of construction) due to enhancement of heat transfer.

7.1.2 Self Cleaning Heat Exchanger

Fouling in the tube-side of the subject thermosyphon reboiler (exchanger case 2) is an ever-lasting problem and its maintenance is very costly. At this point, it is decided to investigate the available fouling mitigation methods and their suitability to the current problem. Based on the investigation outcome, the self-cleaning heat exchanger technology has been selected to test its effectiveness in mitigating the tube-side fouling in the subject heat exchanger. This new technology has been used and proven successful in many services, however, it has not been used in oil service. A schematic of a self-cleaning heat exchanger is shown in Figure 7.2. In self-cleaning heat exchanger, the prone fluid (crude) that flows upward inside the tubes, is charged with solid steel

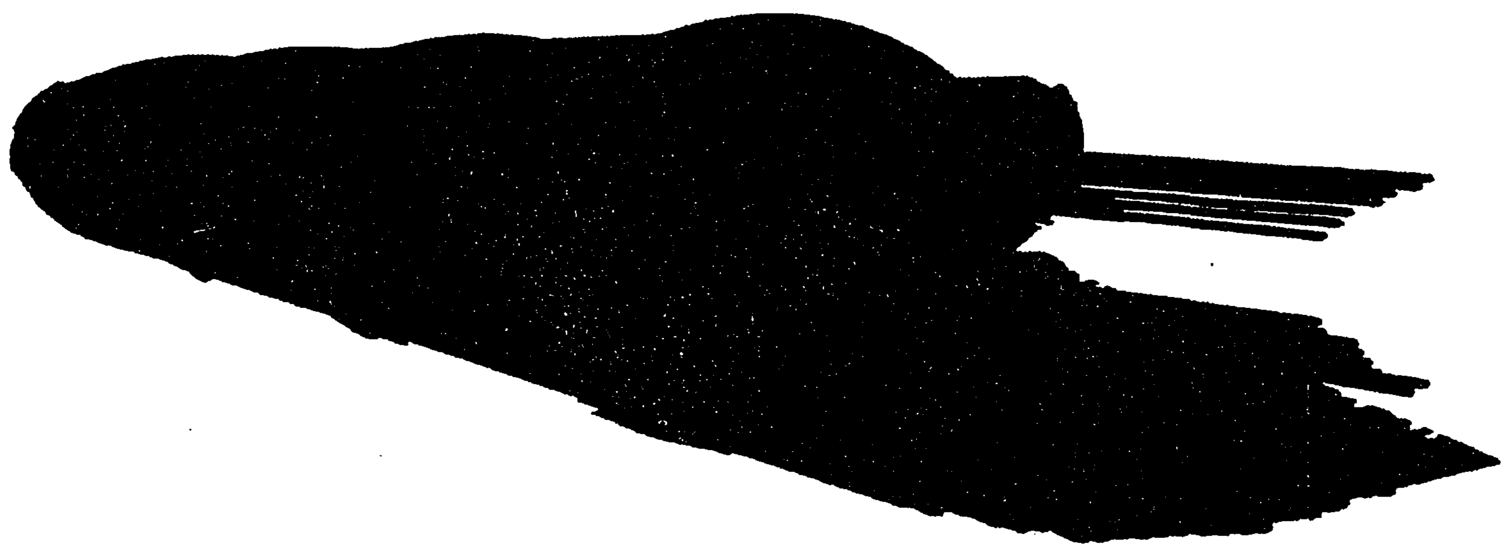


Figure 7.1: Helically Baffled Heat Exchanger

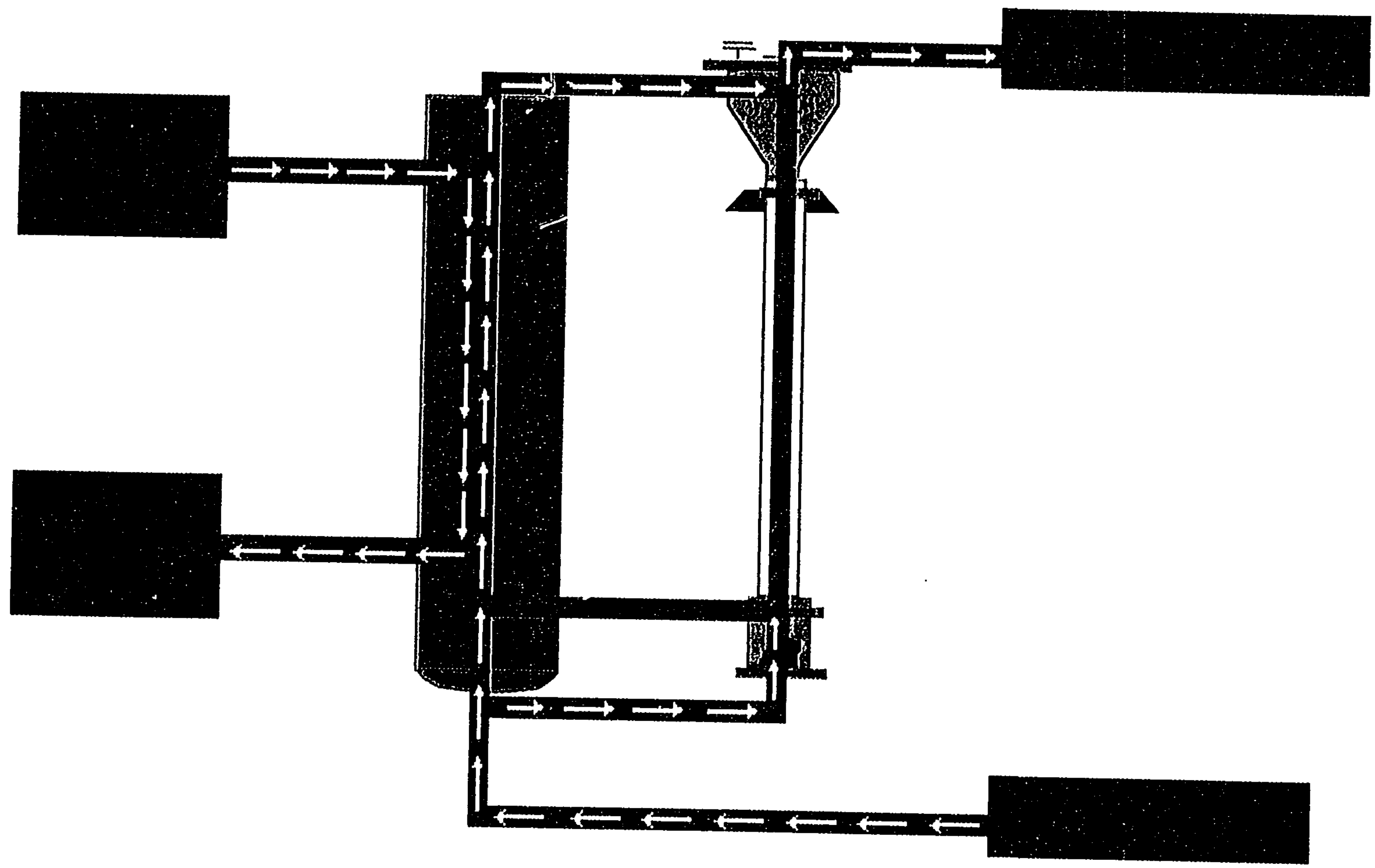


Figure 7.2: Schematic of Self-Cleaning Heat Exchanger

particles (with dimensions typically between 2 to 3 mm) that are swept upward with the crude producing a scouring action on the walls of the tubes. A unique distribution system at the inlet channel provides a uniform distribution of crude and particles into all the tubes. The particles existing from the outlet channel are carried into the separator where they disengage from the crude and are returned through the external downcomer, the control channel and the connecting line into the inlet channel. The flow of particles is activated by the control crude flow, which is a fraction of the total crude flow supplied to the exchanger. By changing the control crude flow, the intensity of the cleaning action can be varied. If desired, the cleaning action can also be applied intermittently. It is expected with this method that there will be no fouling in the exchangers (zero fouling factor) since it could mitigate the fouling completely. In addition to solving the fouling problems, this method will also increase the tube-side heat transfer coefficient of the heat exchanger and thus its efficiencies. This could result in considerable energy savings.

7.2 On Line Experiments and Tested Units

On line experiments were conduced to study new fouling mitigation technologies, namely helically baffled technology for the shell side and self-cleaning technology for the tube side of shell and tube heat exchangers. The helically baffle technology is tested on a naphtha shell and tube heat exchanger installed in the recovery plus plant, whereas, the self-cleaning technology is tested on reboiler installed in the stabilization plant.

7.2.1 Helically Baffled Heat Exchanger

To evaluate the effect of helical baffles on fouling mitigation on the shell side of the naphtha heat exchangers, the naphtha heat exchangers have been converted to helically baffled heat exchanger by replacing the three conventional bundles with helically baffled bundles. The helically baffled heat exchangers designed and sized for the subject heat exchangers. They were designed by ABB Lummas [62] and manufactured by Ohmstede. It took more than 12 months and 2 trips to Netherlands and USA to finalize the design and sign the manufacturing agreement.

The helically baffled bundles used in this study have 12° helical angle and a baffle spacing of 178 mm. The number of tubes in each bundle is 112 U-tubes and the number of tube passes is two. The bundle diameter is 498.5 mm. A picture of the helically baffled bundles is shown in Figure 7.3. The details of the helically baffled bundle are shown in Figure 7.4. This figure shows the plan and elevation views of the bundle. The tube-sheet details of the helically baffled heat exchanger are shown Figure 7.5. This figure shows the locations of the tubes, tie rods, seal rods and baffle rods.

Since the only change for this test is replacing the bundles, the same instruments used in the conventional (original) heat exchangers are used to measure temperatures and flow rates. The temperature and flow transmitters

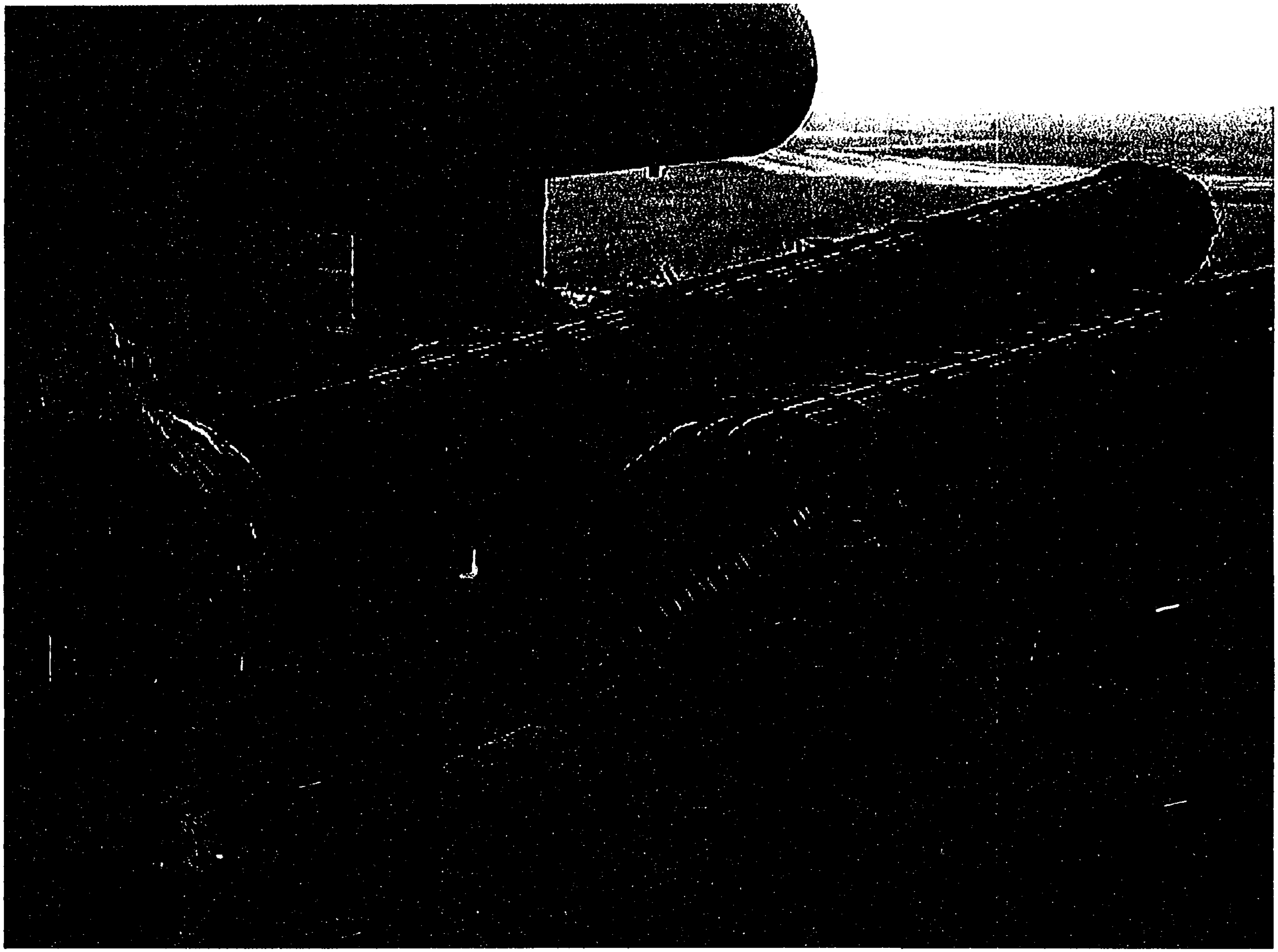


Figure 7.3: Picture of the Helically Baffled Bundles

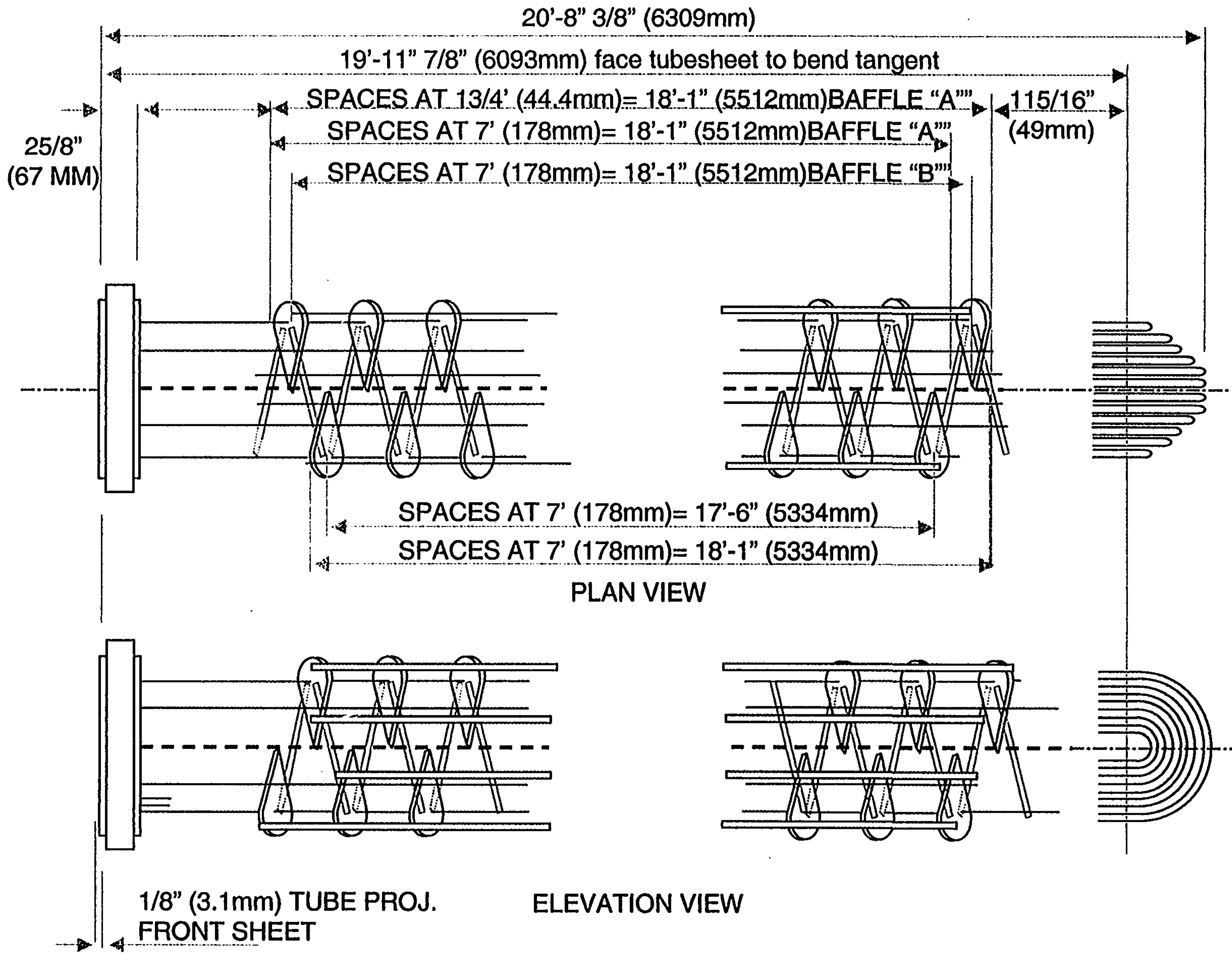


Figure 7.4: Details of the HBHE Bundle

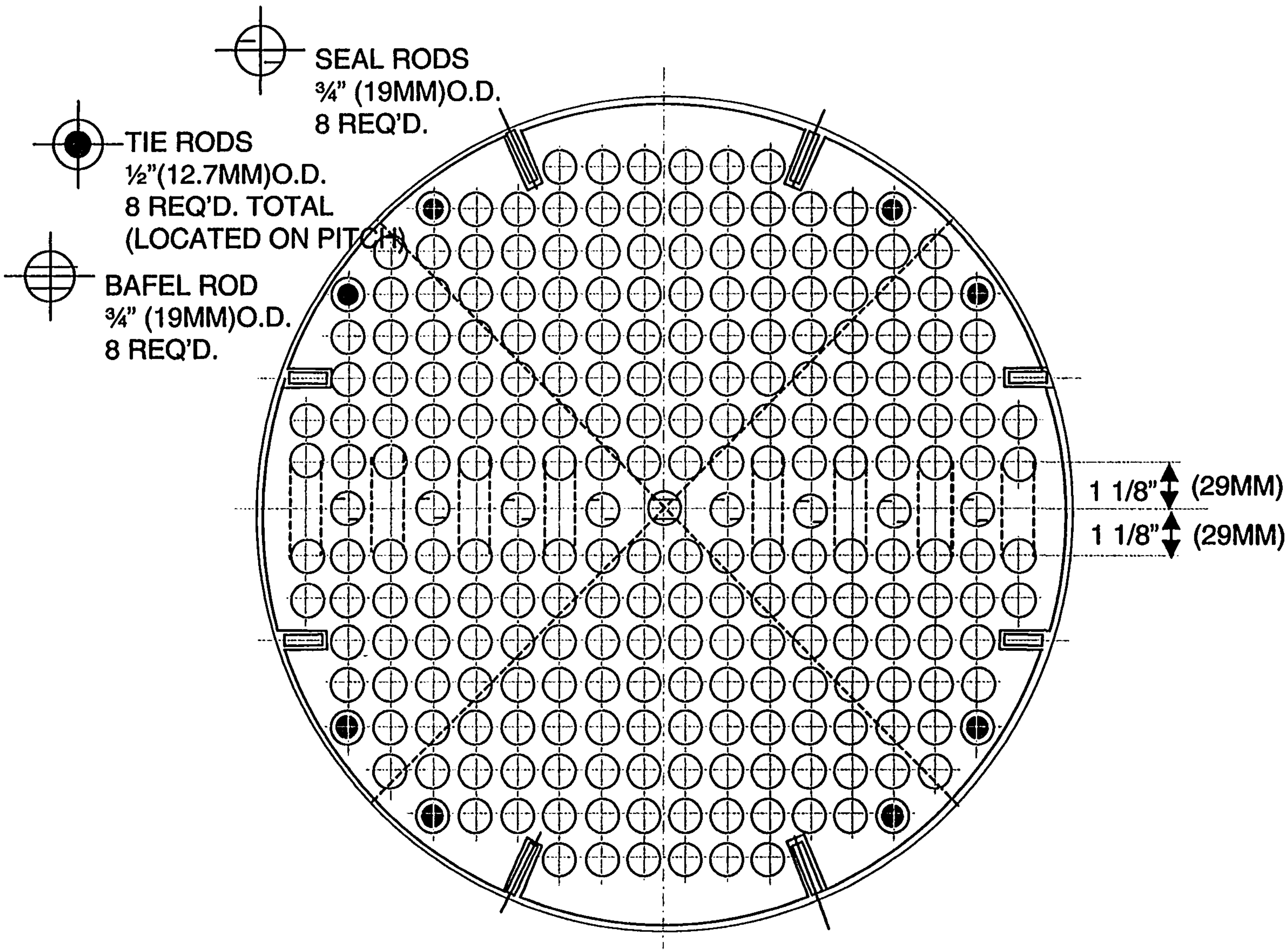


Figure 7.5: View Looking At Shell Side Tubesheet of The HBHE

are connected to the control room for continuous monitoring and data logging. More than 6 months of data were collected after the replacement of the bundles with the helically baffled bundles. The data included inlet and outlet temperatures and mass flow rates for both fluid streams. Samples of these data are shown in Table 7.1 and Appendix A7.

Table 7.1: Sample of the Collected Data for the Modified Heat Exchanger Case 1 (HBHE)

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	3/20/01	41.33	-6.19	-16.54	25.06	15.15	14.67
2	3/21/01	40.67	-6.83	-17.25	24.48	15.10	14.67
3	3/22/01	41.05	-6.93	-17.27	24.55	14.95	14.42
4	3/23/01	39.83	-6.97	-17.24	23.84	14.85	14.42
5	3/24/01	41.81	-6.84	-17.26	24.90	15.04	14.42
1	3/20/01	41.33	-6.19	-16.54	25.06	15.15	14.67

7.2.2 Self Cleaning Heat Exchanger

To evaluate the effectiveness of self-cleaning technology on the mitigation of tube-side fouling in the reboiler, a self-cleaning test unit was designed, manufactured and installed in parallel to the thermosyphon reboiler. It took more than 15 months and 3 trips to Netherlands and USA to finalize the design and sign the contract. The manufacturing of the test unit was done by Klerax Technology BV [54]. Pictures of the self-cleaning heat exchanger test unit are shown in figure 7.6a and 7.6b.

The schematic of self-cleaning heat exchanger test unit is shown in Figure 7.7. As can be seen, the crude (fouling service) enters the test unit at

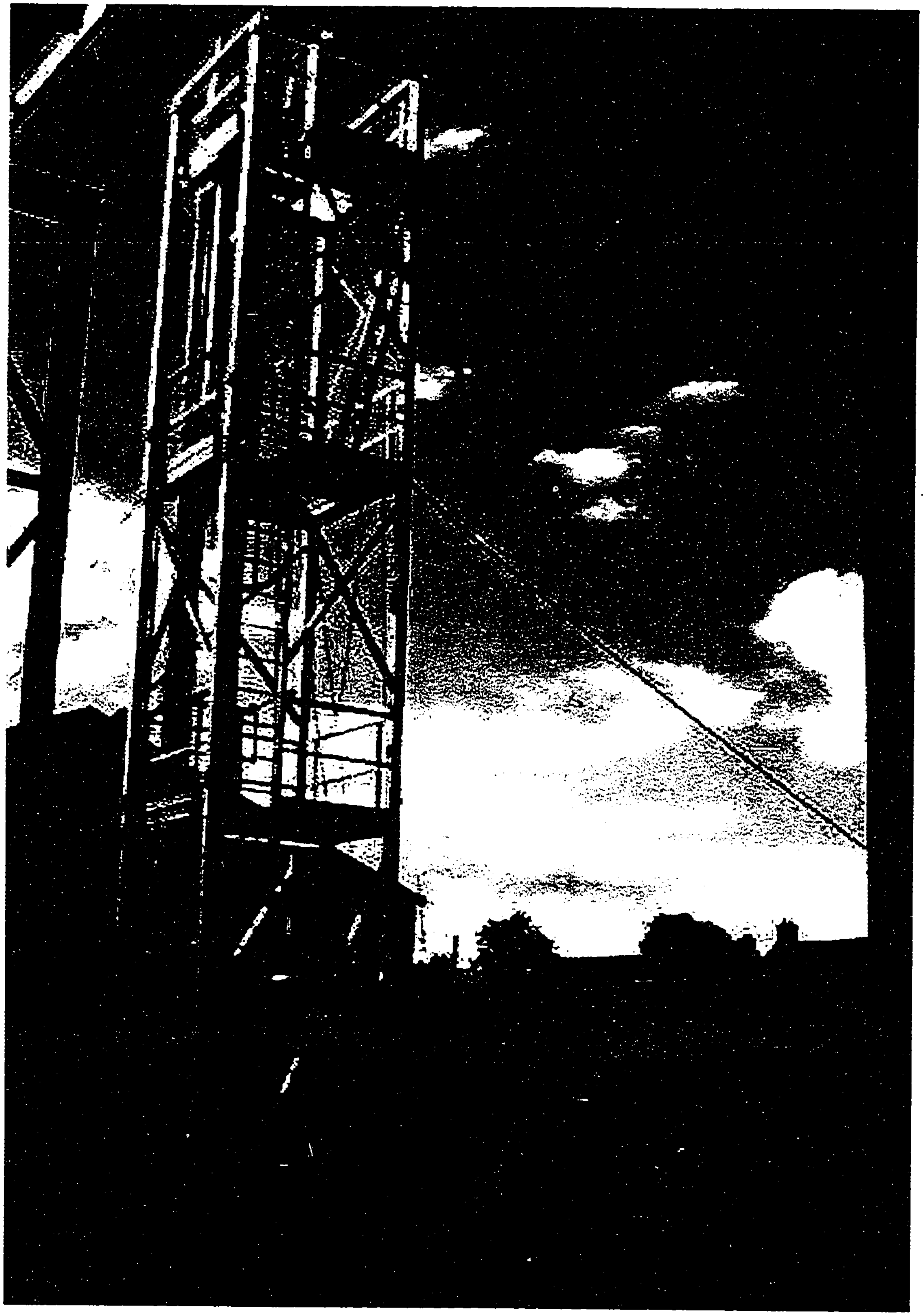


Figure 7.6a: Picture of SCHE Test Unit

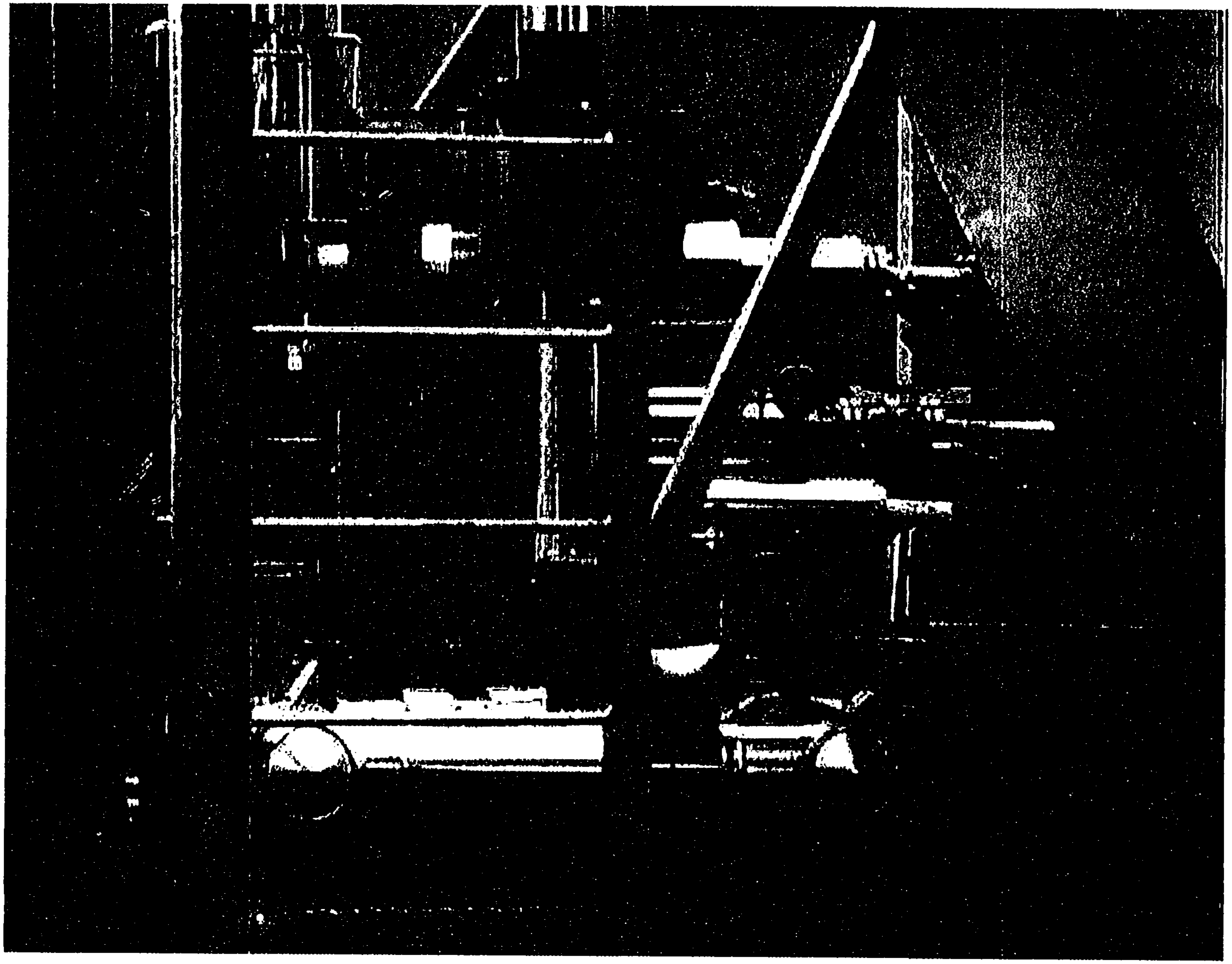


Figure 7.6b: Closer Picture of SCHE Test Unit

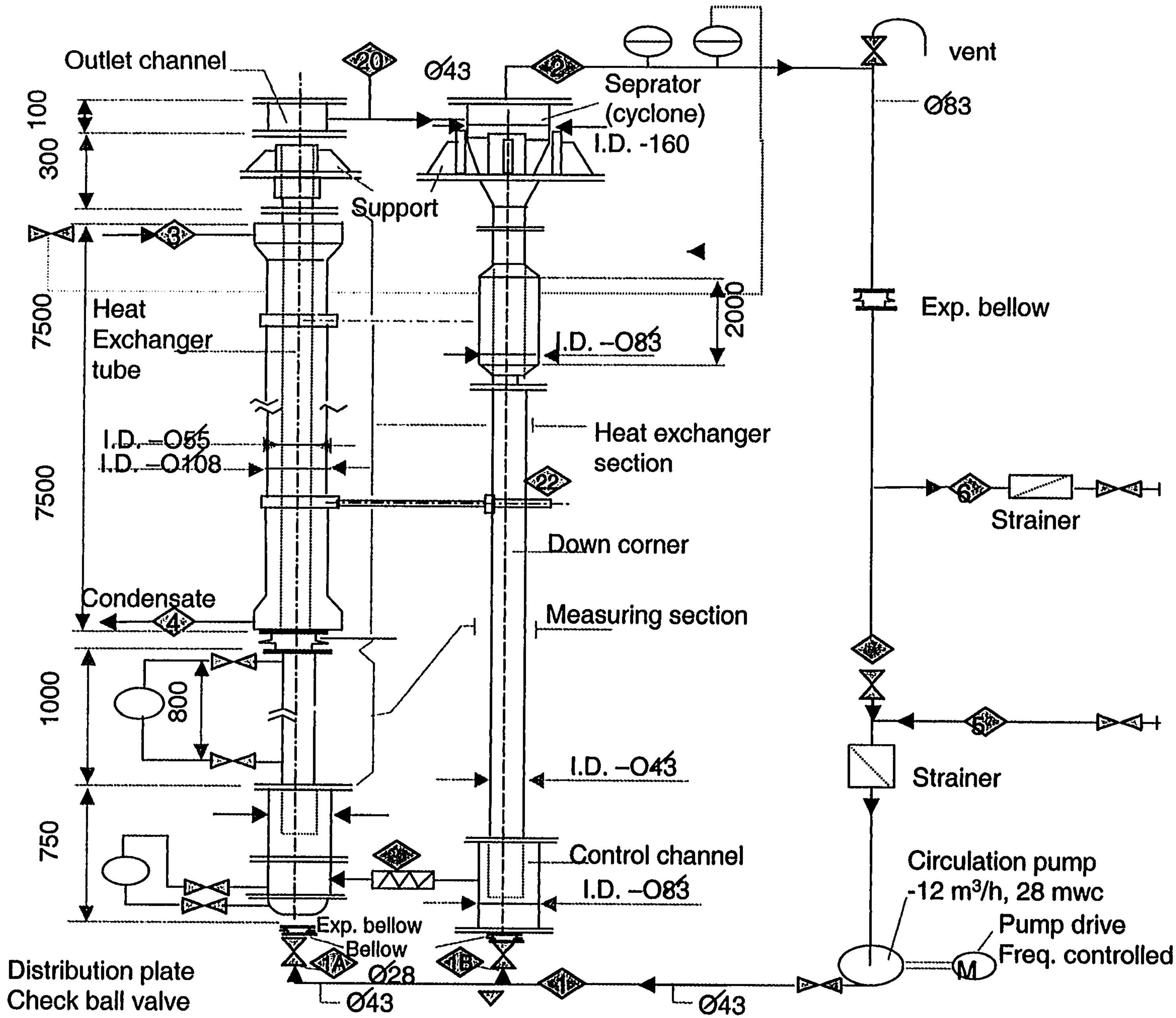


Figure 7.7: Schematic of Self-Cleaning Heat Exchanger Test Unit

line 5, passes through a strainer and is transported by a circulation pump into the exchanger (line 1). The heat exchanger body consists of a vertical double tube heat exchanger with a heat transfer area of 1.42 m^2. The crude, which normally causes severe fouling of heat exchanger surface, is supplied to the inlet channel through line 1A, where it is mixed with the particles, which are supplied from the control channel through line 21. The particles in the tubes are carried by the upward flow and have a scraping effect on the wall of the exchanger tube, which removes any kind of deposit at an early stage. The crude and particles flow from the outlet channel into the separator where the particles disengage from the crude and return through the downcomer to the control channel and again to the inlet channel. The volume of particles carried by the crude flow may vary from 0% to 15% and affects the cleaning action and the value of film coefficient. The flow leaves the exchanger through line 2 and then leaves the test unit through line 6.

For the self-cleaning heat exchanger (SCHE) test unit, the instruments and equipment used are temperature transmitters, flow meters, differential pressure transmitters, data logger and pressurized cabinet.

The test unit has temperature transmitters [95] at the inlet and outlet for both the shell and tube sides. The temperature transmitters are RTD type with a temperature range from –50 ° C to 250° C and inaccuracy of 0.2%. However, an inaccuracy of 0.3% is reported in the instrument engineers' handbook.

The flow meter transmitter [96] used in the SCHE test unit is differential pressure flow nozzle type. The flow rate range is from 0 m^3/hr to 12 m^3/hr with inaccuracy of about 2.0% which is higher than the value (1.5%) reported in the instrument engineers' handbook [74].

The differential pressure transmitters [97], as shown in Figure 7.8, are used to measure the pressure drop across one-meter length of the tube. The collected pressure drop data across the one-meter tube length is used to estimate the porosity required for the thermal analysis of the self-cleaning heat exchanger. The manufacturer developed a relationship between the porosity and the pressure drop across the one-meter tube length (i.e for several known porosities, the pressure drop across the one-meter length was measured and then the regression correlation is developed). The differential pressure range is from 497 pa to 6895 kpa with inaccuracy of about 2.0% which is higher than the value (1.5%) reported in the instrument engineers' handbook [74].

The collected data in the SCHE test unit is monitored and stored in the data logger called Skipper [73]. The Skipper is a versatile data logger, which by means of connected sensors, measures quantities such as temperatures, flow rates, etc. and stores the data on an exchangeable memory card. The processing consists of reading-in the measurement data with a card reader. Then the data can be stored. Sample of the collected data shown in Table 7.2 and Appendix B7.

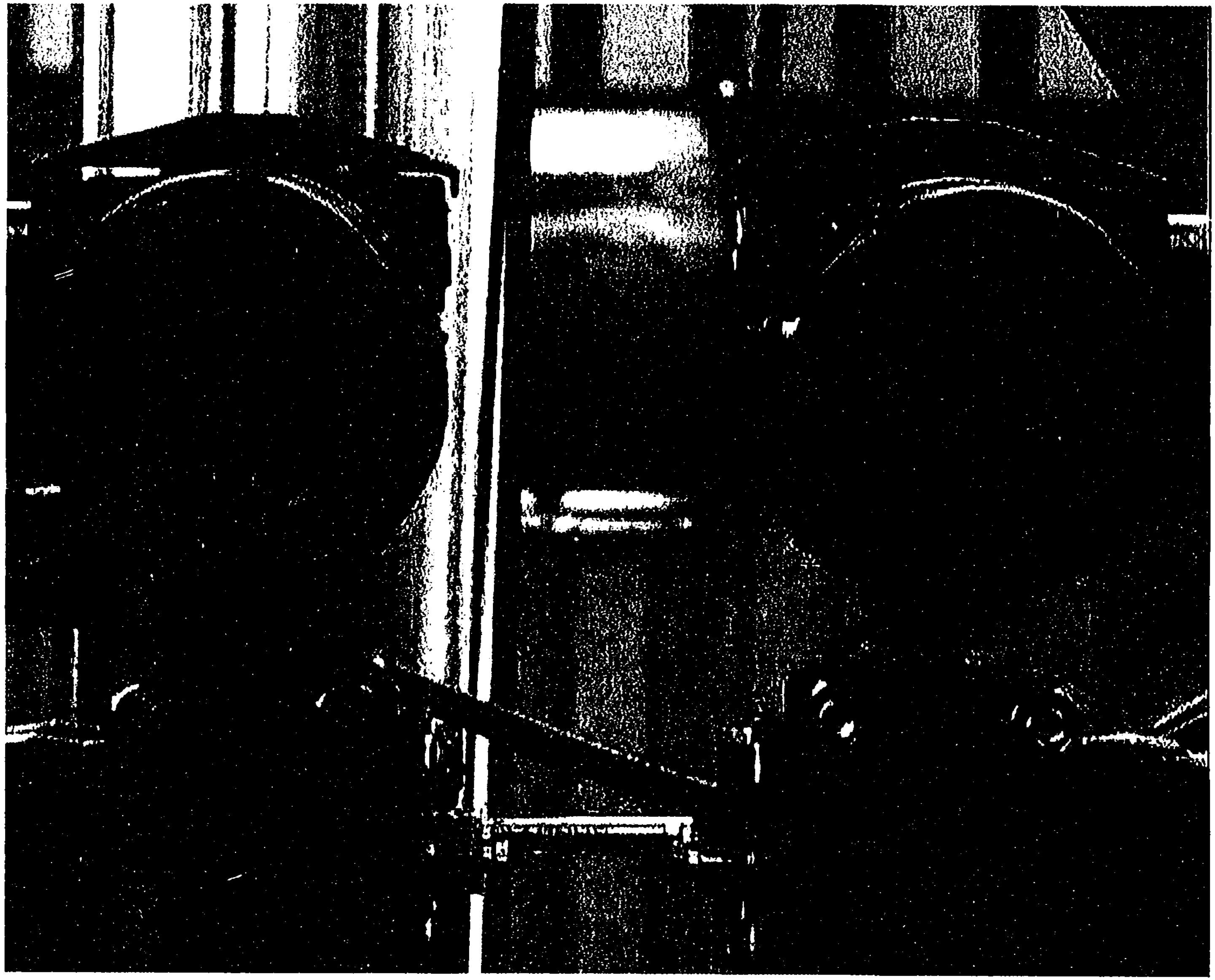


Figure 7.8: Pressure Transmitters

Table 7.2: Sample of the Thermal Data for SCHE Test Unit

Day	Date	Total Flow (m^3/hr)	Control Flow (m^3/hr)	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)
1	6/20/00	5.30	1.07	100.0	60.7	55.0	64.3
6	6/26/00	5.20	1.14	100.0	60.7	57.5	65.2
10	6/30/00	5.00	0.82	100.0	61.8	56.8	66.0
14	7/4/00	5.30	0.90	100.0	61.3	56.7	64.9
20	7/10/00	4.80	0.70	100.0	63.0	53.7	65.0

Since the SCHE test unit is located within a hazard area (area contain flammable hydrocarbon), then the data logger should be stored in a pressurized cabinet as recommended by the plant loss prevention department. A picture of the cabinet, which contains the data logger, is shown in Figure 7.9. An explosion proof pressurized cabinet [99] is purchased and installed next to the SCHE test unit. Air is flowing to the cabinet to maintain an over-pressure of about 80 pa in the cabinet. The pressure inside the cabinet is monitored continuously by a pressure switch.

7.3 Equations of Heat Transfer Coefficients and Pressure Drop

The same equations used to perform thermal analysis are used to evaluate the effectiveness of the new mitigation technology. However, there are some changes or additions for the shell side of the helically baffled heat exchanger and the tube side of the self-cleaning heat exchanger.

For the helically baffled heat exchanger, a new correction factor that accounts for the helical baffle configuration to determine the shell side heat

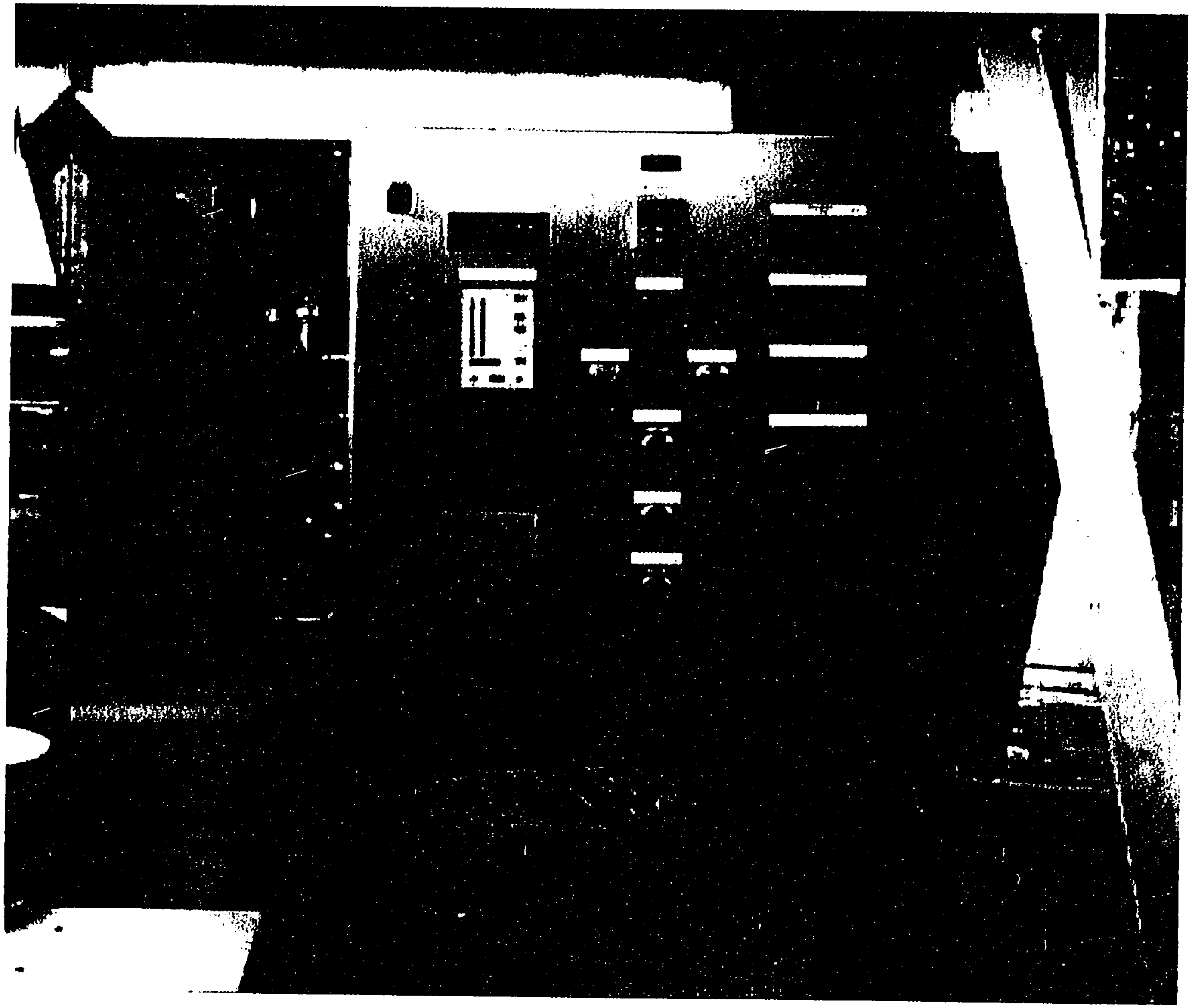


Figure 7.9: Pressurized Cabinet and Data Logger

transfer coefficient is introduced. The equation of this factor is discussed in the results and discussion section

For the self cleaning heat exchanger (SCHE), the tube side heat transfer coefficient developed by Ruckenstenin [55] is given as follow:

$$h_t = \frac{k_t}{d_p} 0.067 \ \mathrm{Pr}_t^{0.33} \ \mathrm{Re}_{dp}^{-0.237} \ Ar^{0.522} \quad if \ \ \mathrm{Re}_{dp} \ \ Ar^{-0.58} > 0.09 \qquad (7.1)$$

or

$$h_t = \frac{k_t}{d_p} 0.326 \ \mathrm{Pr}_t^{0.33} \ \mathrm{Re}_{dp}^{0.423} \ Ar^{0.14} \quad if \ \ \mathrm{Re}_{dp} \ \ Ar^{-0.58} < 0.09 \qquad (7.2)$$

Where:

d_p : *Particle diameter, m*

$$\mathrm{Re}_{dp} = \frac{\rho_t V_{1,s} d_p}{\mu_t}$$

$$Ar: \ Archimedes\ number = \frac{\rho_t^2 g \ d_p^3}{\mu_t^2} \frac{(\rho_s - \rho_t)}{\rho_t}$$

$V_{1,s}$: *Superficial liquid velocity in tubes* $= e V_t$

Where:

e : Porosity = Volume fraction of liquid in the tube, dimensionless

= 1- volume fraction of particles in the tube.

Also, the tube side pressure drop for the SCHE, Δp_{SCHE} , is given as:

$$\Delta P_{SCHE} = K_{SC} \ \Delta P \qquad (7.3)$$

Where ΔP is the tube-side pressure drop without particles and is given as:

$$\Delta p = \frac{4L}{D_i} \frac{f}{2} \rho_t V_t^2 \tag{7.4}$$

f is the friction factor and is given as [25]:

$$f = (1.58 \ln \mathrm{Re}_t - 3.28)^{-2} \tag{7.5}$$

And K_{SC} is a new added factor (pressure drop factor) introduced in this study to account for the particle in the tube. In this study, after several trials and errors for the best model, the following model for the pressure drop factor is suggested:

$$K_{SC} = 39.7452 \times 10^6 \left(e^{4.0} \mathrm{Re}_{SC}\right)^{-1.7147} \tag{7.6}$$

Where:

Re_{SC} : The Modified Reynolds number $= \dfrac{\rho_m V_t D_i}{\mu}$

ρ_m : Modified density, kg/m^3 $\rho_m = e\,\rho_t + (1-e)\,\rho_s$

ρ_s · Particle density, kg/m^3

7.4 Results and Discussions

The results and discussions for the tested helically baffled heat exchanger and self-cleaning heat exchangers are given in the following sections:

7.4.1 Helically Baffled Heat Exchanger

This set of results pertains to the evaluation of the procedure adapted to mitigate/reduce the shell side fouling problem in the subject heat exchanger. The bundle of the subject exchanger is replaced by a helically baffled heat exchanger (HBHE). A Sample of the operational data for the modified heat exchanger (HBHE) is shown earlier in Table 7.1 and Table A7.

Thermal analysis method has been used to determine fouling growth. Such analysis is not reported in the literature. Hence, to determine the corrected heat transfer coefficient on the helically baffled shell side, a new correction factor (J_H) that accounts for the helical configuration has been developed using the first two months data. By assuming that the heat exchanger is clean during the first two months (which was confirmed by visual inspection), the actual shell side heat transfer coefficient and hence the helical correction factor J_H could be determined using regression and the collected data as shown in Table 7.3 and Table A8.

Table 7.3: Results for Exchanger Case 1 to Develop J_H Factor

Day	Re_s	h_c (W/m^2 K)	J_C	J_L	J_B	h_s (W/m^2 K)	Re_t
1	9462.95	1179.54	1.02	0.86	0.93	962.34	14854.20
2	9520.41	1184.20	1.02	0.86	0.93	966.14	14893.85
3	9371.01	1172.07	1.02	0.86	0.93	956.25	14596.24
4	9307.78	1166.91	1.02	0.86	0.93	952.04	14596.61
5	9428.60	1176.75	1.02	0.86	0.93	960.07	14597.47

Table 7.3: Results for Exchanger Case 1 to Develop J_H Factor (cont.)

Day	h_t (W/m^2 K)	R_k (W/m^2 K)	U_a (W/m^2 K)	$\dot{Q}$ (W)	LMTD (°C)	F	J_H
1	1102.74	5.0E-05	496.68	1292831	13.09	0.73	1.30
2	1105.10	5.0E-05	497.55	1296433	13.09	0.73	1.30
3	1087.40	5.0E-05	489.25	1282858	13.18	0.73	1.29
4	1087.42	5.0E-05	486.15	1251692	12.92	0.73	1.27
5	1087.47	5.0E-05	485.14	1301093	13.40	0.73	1.25

Figure 7.10 shows the variation of the helical correction factor J_H with Reynolds number (Re) and the best-fit curve with 3rd degree polynomial. These fits have both high coefficient of determinations and adjusted coefficient of determinations. Since the increase in the coefficient of determination is very small with polynomials higher than the 3rd degree, the 3rd degree polynomial is selected to represent the helical correction factor J_H that is found to be:

$$J_H = -2.472 \times 10\text{-}^{10}\, Re^3 + 7.535 \times 10^{6}\, Re^2 - 7.661 \times 10^{2} Re + 2.609 \times 10^{2}$$

The developed helical correction factor J_H is used to determine the corrected heat transfer coefficient on the shell side of the HBHE and hence the overall heat transfer coefficient. Sample of the detailed results are shown in Table 7.4 and Table A10.

Table 7.4: Sample of Results for Exchanger Case 1 after HBHE Installation

Day	U_a (W/m^2 K)	h_t (W/m^2 K)	h_c (W/m^2 K)	J_H	h_s (W/m^2 K)	U_c (W/m^2 K)	Fouling Resistance (hr. ft^2 F/BTU)
1	496.38	1102.74	1182.15	1.24	1191.60	486.88	0.0000
2	496.71	1102.74	1179.56	1.25	1199.46	488.19	0.0000
3	489.37	1087.40	1172.06	1.27	1217.88	487.28	0.0000
4	486.02	1087.42	1166.88	1.29	1224.48	488.34	0.0001
5	485.19	1087.47	1176.81	1.26	1207.11	485.56	0.0000

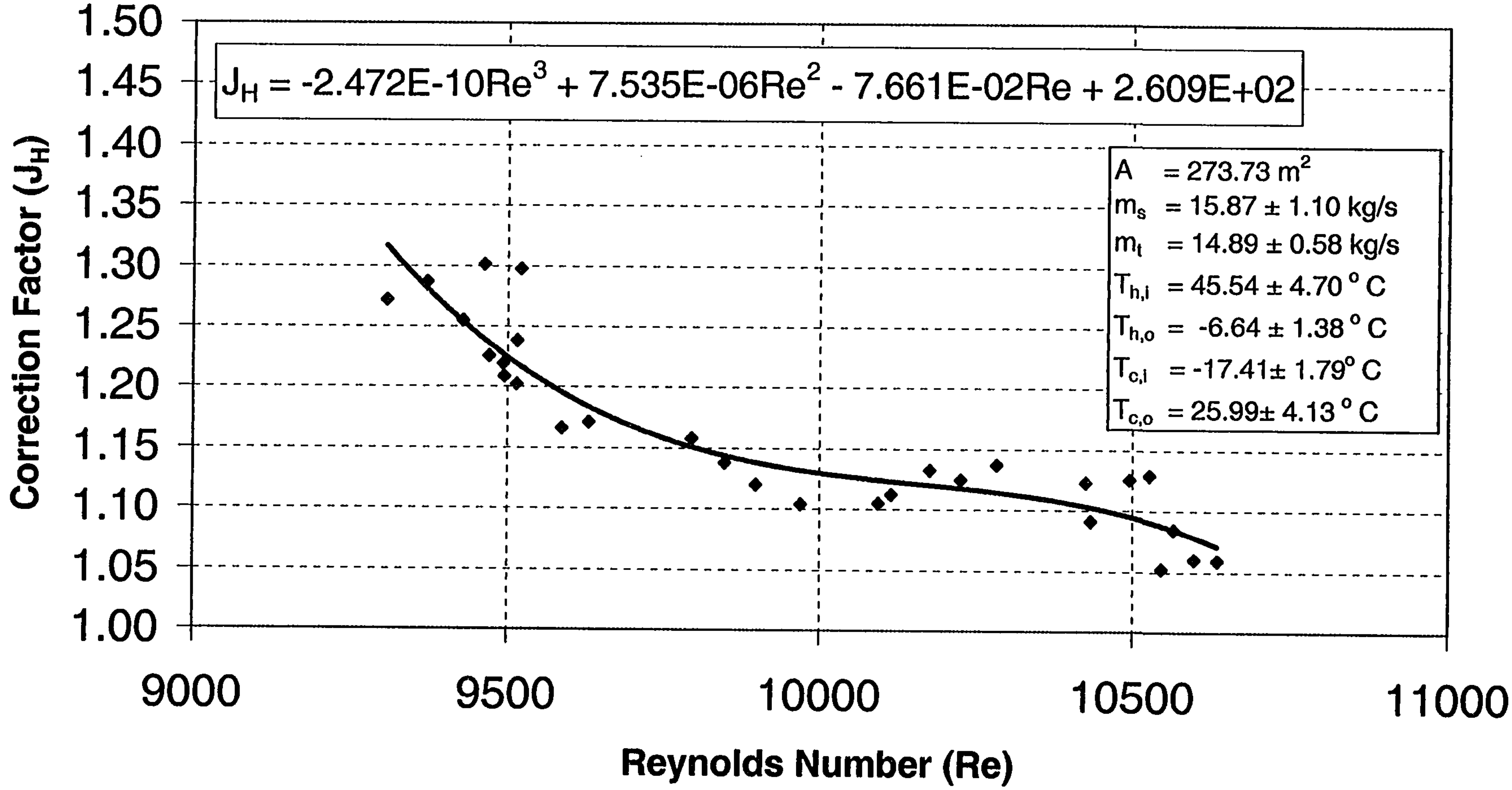


Figure 7.10: Variation of the Helical Correction Factor J_H with Reynolds Number Re (3rd Degree Polynomial)

Figure 7.11 shows the clean and actual overall heat transfer coefficient for more than 6 months duration. The reduction in the actual overall heat transfer coefficient is very small as compared to results for the ordinary shell and tube heat exchanger. Figure 7.12 shows the fouling growth for both the HBHE and the ordinary shell and tube heat exchanger for a period of four months. As can be seen from the figure, introducing helical baffles on the shell side of a shell and tube heat exchanger reduces significantly the fouling growth.

7.4.2 Self Cleaning Heat Exchanger

This set of results pertains to fouling mitigation in the tube side of shell and tube heat exchangers. The fouling mitigation technology that has been used for the heat exchanger case 2 is self-cleaning. Such technology reduces fouling but increases pressure drop. The geometrical and installation details of the self-cleaning test unit is shown and described in section 7.2. The test unit is instrumented to measure the tube-side pressure drop along one-meter length of the self-cleaned section. Table 7.5 and Tble B6 give sample of the measured data and the calculated parameters and results.

Table 7.5: Data and Results for Hydraulic Analysis of SCHE Test Unit

Day	Total Flow (m^3/hr)	Control Flow (m^3/hr)	Porosity e	Pressure Drop (with Particles) Pa	Pressure Drop (W/O Particle) Pa	Velocity (m/s)	Re_{sc}
1	9.0	0.45	0.950	2000	288.3	1.06	11716.8
2	9.0	0.60	0.935	3500	288.3	1.06	7605.9
3	9.0	0.70	0.925	4000	288.3	1.06	7524.6
4	9.0	0.80	0.920	4700	288.3	1.06	7483.9
5	9.0	0.90	0.915	4900	288.3	1.06	7443.2

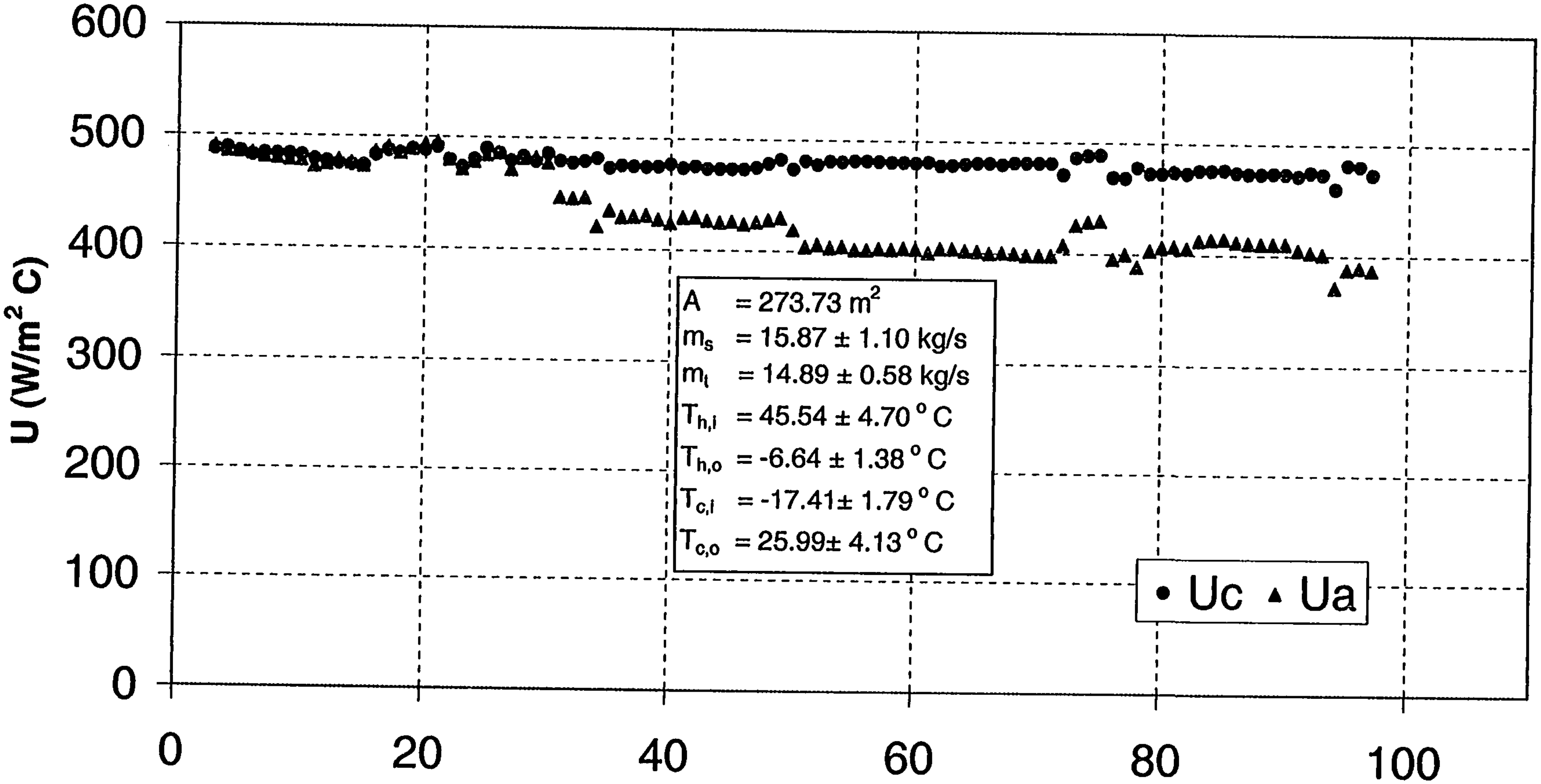


Figure 7.11: Actual and Clean Overall Heat Transfer Coefficients Versus Time for the Helically Baffled Heat Exchanger

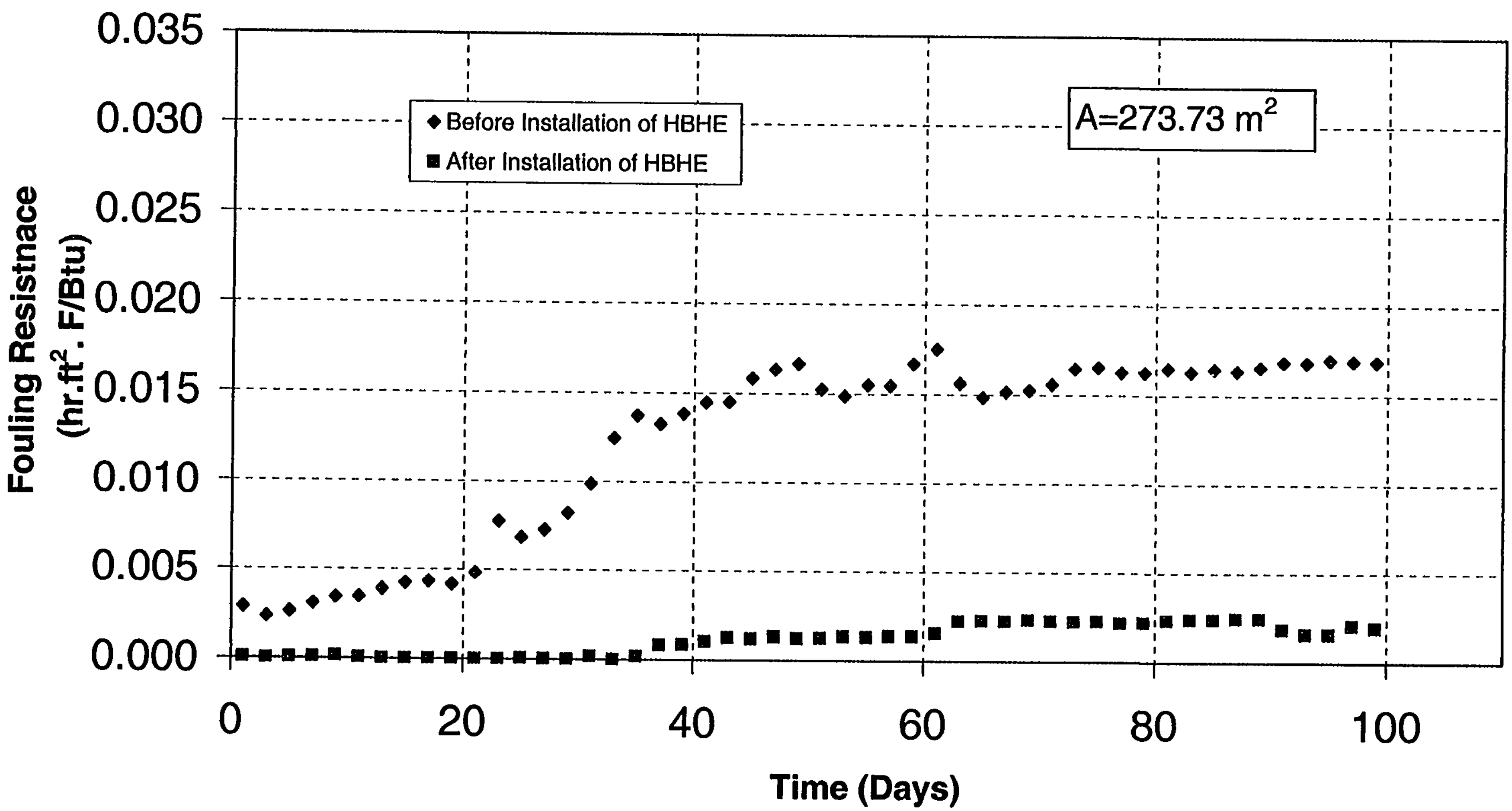


Figure 7.12: Fouling Growth Versus Time Before and After the installation of the Helically Baffled Heat Exchanger

Table 7.5: Data and Results for Hydraulic Analysis of SCHE Test Unit (Cont.)

Day	Friction Factor f	Pressure drop Factor K_{sc}	Ln (K_{sc})	Ln (Re_{sc} e^4)	Calculated Pressure Drop (with Particles) Pa
1	0.00753	5.34	1.675	9.164	2232.5
2	0.00851	12.73	2.544	8.668	3909.3
3	0.00854	14.66	2.685	8.614	4329.6
4	0.00855	17.30	2.851	8.587	4558.3
5	0.00856	18.10	2.896	8.560	4800.4

The data has been used to develop a correction for a newly suggested pressure drop factor K_{sc} that accounts for the cleaning particles and porosity in the tube side. Using the data shown in Figure 7.13, curve fitting and the calculation procedure described in section 7.3, the developed pressure drop factor for SCHE is found to be:

$$K_{SC} = 39.7452 \times 10^{6} \left(e^{4.0} \operatorname{Re}_{SC}\right)^{-1.7147}$$

Where the porosity (e) as a function of the total flow in m^3/hr (as X_1) and the control flow rate in m^3/hr (as X_2) is found to be:

$$e = 0.9725 + 0.0142X_1 - 0.375X_2 + 0.2193X_2^2 - 0.0388X_2^3$$

Samples of measured and calculated parameters required to calculate the fouling growth in the self cleaning heat exchanger test unit are given Tables 7.6, 7.7, B7 and B8.

Table 7.6: Sample of the collected Data for SCHE Test Unit

Day	Date	Total Flow (m^3/hr)	Control Flow (m^3/hr)	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)
1	6/20/00	5.30	1.07	100.0	60.7	55.0	64.3
7	6/26/00	5.20	1.14	100.0	60.7	57.5	65.2
11	6/30/00	5.00	0.82	100.0	61.8	56.8	66.0
15	7/4/00	5.30	0.90	100.0	61.3	56.7	64.9
21	7/10/00	4.80	0.70	100.0	63.0	53.7	65.0

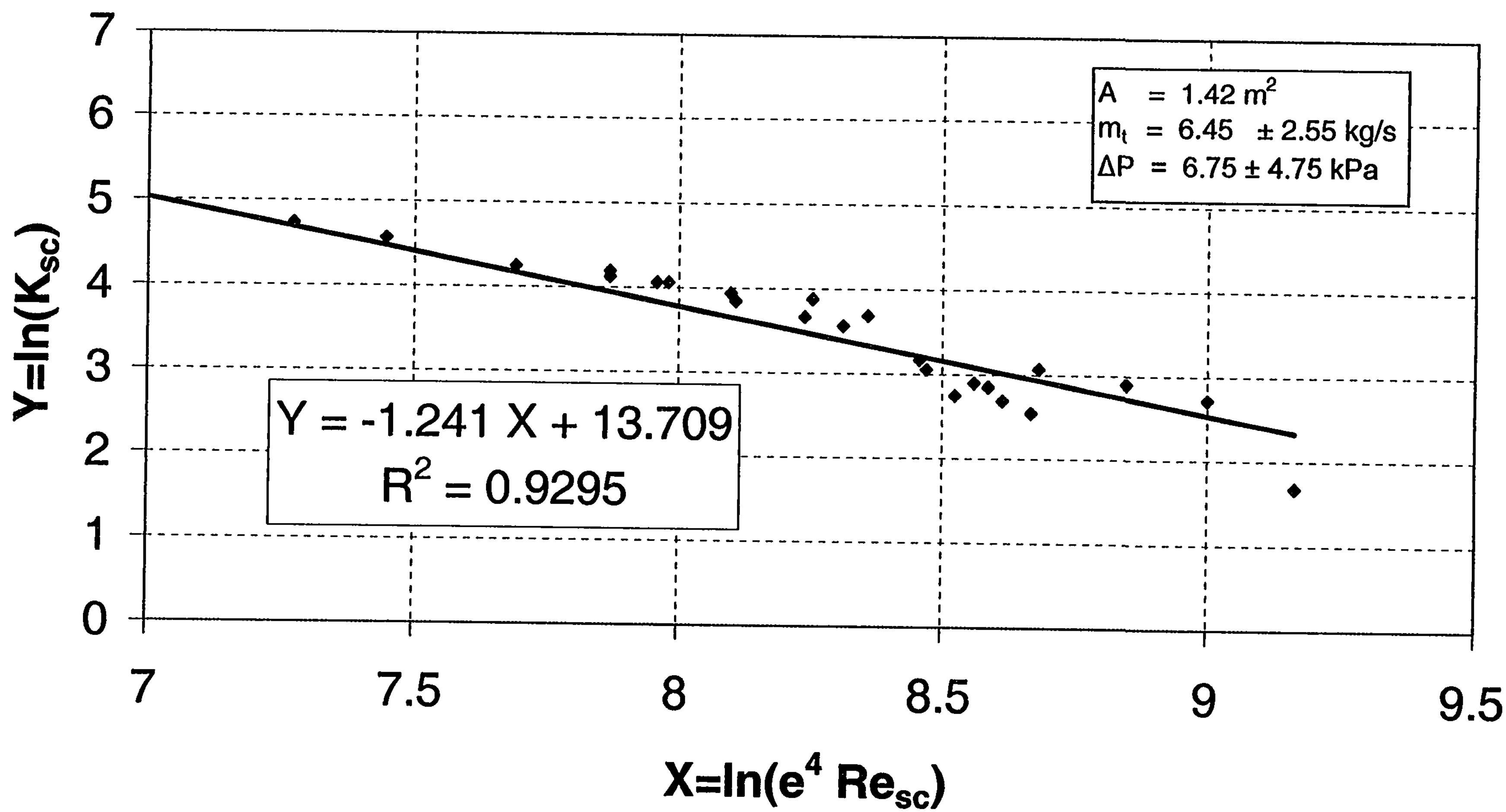


Figure 7.13: Pressure Drop Factor K_{sc}

Table 7.7: Sample of the Thermal Results for SCHE Test Unit

Day	Porosity e	$\dot{Q}$ (W)	LMTD (°C)	U_s (W/m² K)	Vt (m/s)	Re_{sc}	Re_t
1	0.850	22746.8	16.4	977.80	0.632	32961.5	13762.3
7	0.846	18478.0	13.2	980.88	0.620	32791.1	13502.6
11	0.862	21228.5	15.1	986.32	0.596	29587.7	12983.3
15	0.859	20056.3	15.0	939.29	0.632	31765.3	13762.3
21	0.872	25031.2	19.4	907.33	0.572	27321.2	12464.0

The correlations developed by Ruckenstenin [55] have been used to calculate the tube side heat transfer coefficient for SCHE. Figure 7.14 shows the calculated tube-side heat transfer coefficients for the test unit with and without particles versus time. As can be seen that the values of SCHE tube-side heat transfer coefficient are 4 to 5 times the values of heat transfer coefficient of the conventional heat exchangers (without particles). This is due to the breakdown of the laminar thermal boundary layer near the tube walls due to the scouring action of particles.

The plot of fouling growth versus time for SCHE is shown in Figure 7.15. Also a comparison between the fouling growth for the SCHE and the conventional one is shown in the same figure. The figure indicates that the fouling growth in the SCHE is almost negligible and the self-cleaning is very effective in mitigating the tube-side fouling.

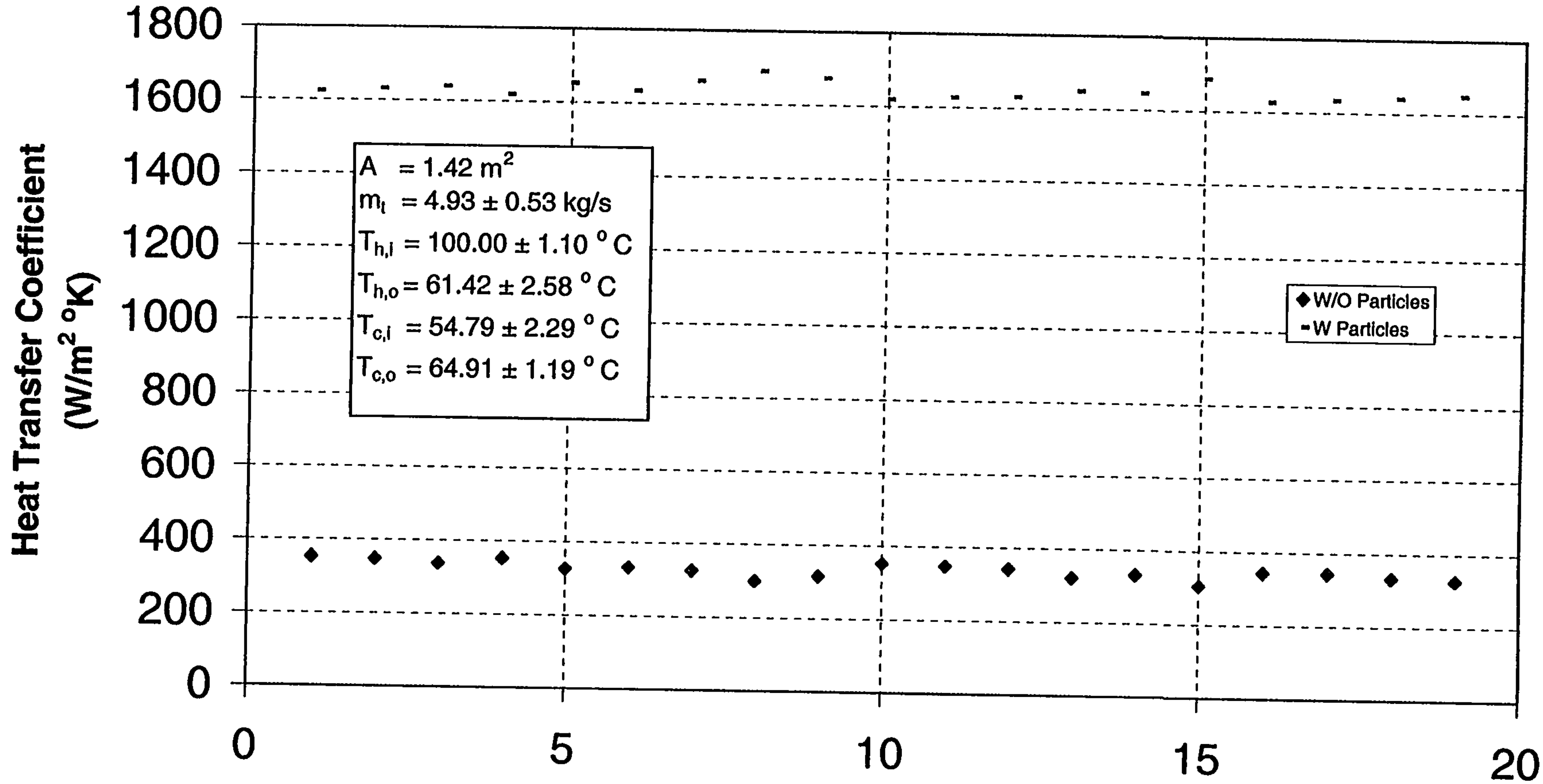


Figure 7.14: Tube Side Heat Transfer Coefficients (With and Without Particles) Versus Time for SCHE Test Unit

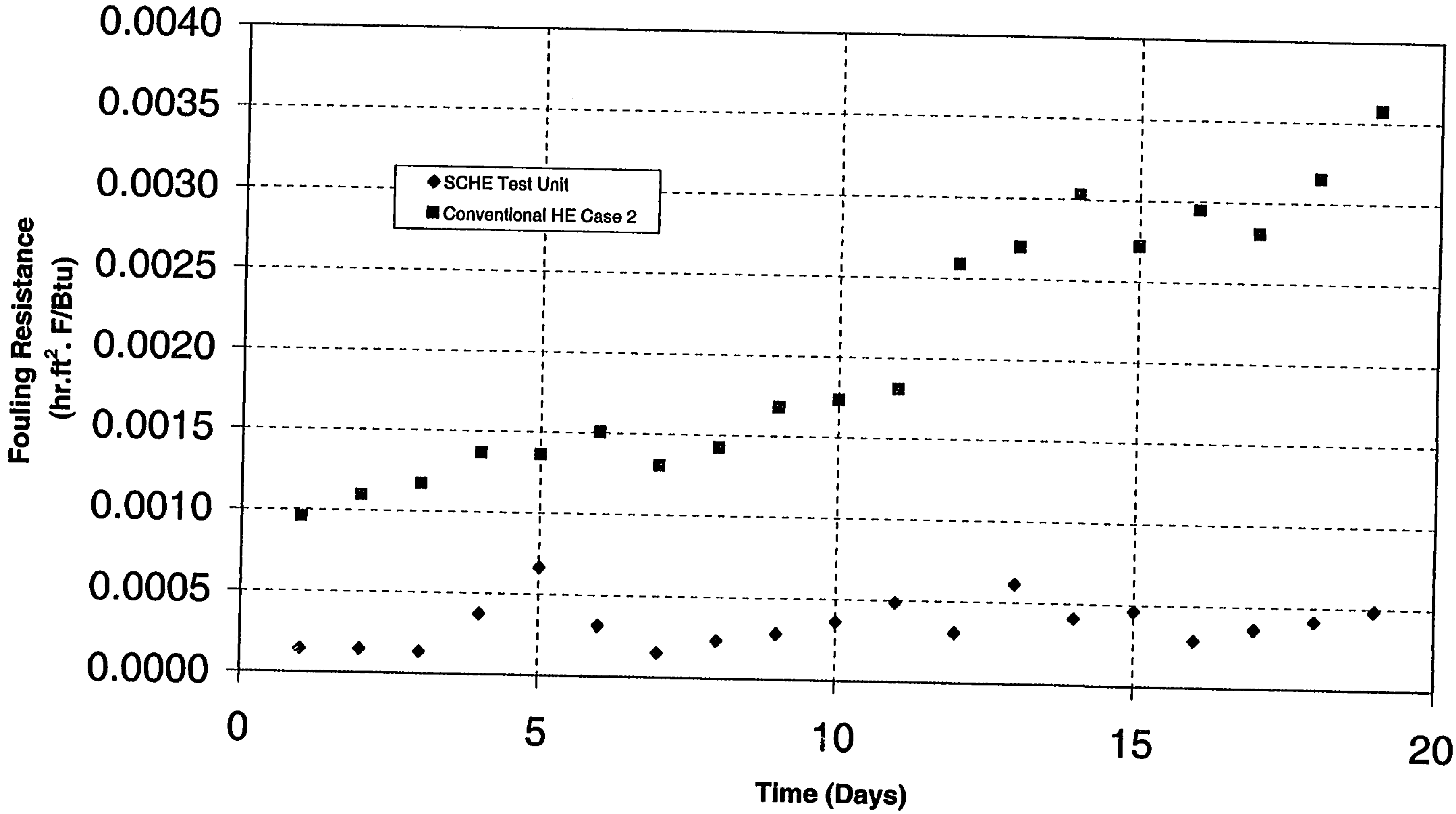


Figure 7.15: Fouling Growth Versus Time for SCHE and Conventional Exchanger Case 2

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

In this study the fouling problem has been analyzed in two types of heat exchangers used in the oil industry using different analysis methods. Also, two mitigation technologies have been recommended, implemented and evaluated. Based on the study results, the following conclusions are made:

- Helical baffles arrangement is very effective in reducing shell side fouling. The performance of the helically baffled heat exchanger indicates a significant reduction in fouling after a six-month operation period.
- The self-cleaning technology is very effective in eliminating tube side fouling. The performance of the self-cleaning heat exchanger test unit indicates negligible fouling growth after a six-month operation period.
- The results of thermal evaluation analysis indicate that there is a difference between the results obtained from Bell Delaware method and Flow Stream Analysis method. Comparison with known commercial software indicates that the Bell Delaware method gives more accurate results.

- A new correction factor (J_H) has been developed for a helically baffled heat exchanger. This factor accounts for the helical baffle configuration to determine the corrected heat transfer coefficient on the shell side.
- A new pressure drop correction factor (K_{sc}) has been developed for a self-cleaning heat exchanger. This factor accounts for the cleaning particles and porosity in the tubes of a self-cleaning heat exchanger.
- A relationship between irreversibility rate and fouling growth in heat exchangers has been developed. This relation indicates that irreversibility rate increases as fouling increases.
- Statistical analysis using a collection of sample function of fouling growth curves can be used to predict the average time required to reach to the critical level of fouling and its variance. By utilizing the underlying link between fouling growth laws and time to reach a critical level of fouling distribution parameters, appropriate heat exchanger maintenance policies can be developed.

8.2 Recommendations for Future Work

The following has been recommended as an extension for this study:

(i) Consider the exchanger performance when the self-cleaning technology is combined with helically baffled configuration.

(ii) Data measurements should be continued for a long period (2 years). Such long-term data should be analyzed to validate the newly developed heat transfer correction factor (J_H) for the helically baffled arrangement and the pressure drop correction factor (K_{sc}) for the self-cleaning arrangement.

(iii) Develop maintenance strategies for cleaning schedules of exchangers case 1 and case 2.

(iv) Perform detailed economical analysis for the use of helically baffled and self-cleaning heat exchangers.

Appendix A

Data and Sample Results for Exchanger Case 1

Table A1: Construction, Process, and Thermal Data for Exchanger Case 1

Shell internal diameter D_s (m)	0.498	
Number of tubes N_T	250	
Tube outside diameter D_o (m)	0.01905	
Tube inside diameter D_i (m)	0.01483	
Tube pitch P_T (m)	0.0254	
Baffle Spacing L_B (m)	0.381	
Shell Length L_S (m)	18.288	
Tube to baffle diametral clearance Δ_{tb} (m)	0.001	
Shell to baffle diametral clearance Δ_{sb} (m)	0.005	
Bundle to shell diameteral clearance Δ_b (m)	0.035	
Number of sealing strips per cross flow Nss/Nc	0.200	
Thickness of baffles t_b (m)	0.005	
Number of tube-side passes n	2	
Baffle cut	25%	
L_c (m)	0.125	
	Shell Side (Naphtha)	**Tube Side (Naphtha)**
Density (kg/m^3)	752	800
Thermal conductivity (W/(m.K))	0.1332	0.1332
Specific heat capacity (J/(Kg. K))	2051.63	1854.8
Viscosity (N.S/m^2)	0.000546	0.000678
Prandtle Number Pr	8.4098347	9.441099

Table A2: Samples of the Operation Data for Exchanger Case 1

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	4/19/99	41.83	-5.71	-17.65	24.71	10.47	9.70
3	4/21/99	43.36	-5.85	-17.69	24.27	11.90	10.55
5	4/23/99	41.04	-5.62	-17.72	23.35	11.80	10.80
7	4/25/99	44.00	-5.95	-17.71	22.79	12.81	10.80
9	4/27/99	43.50	-5.98	-17.67	22.66	12.75	10.80
11	4/29/99	42.52	-6.07	-17.79	21.90	11.84	10.06
13	5/1/99	42.78	-5.84	-17.74	22.22	11.36	9.71
15	5/3/99	44.47	-5.98	-17.88	22.22	11.74	9.70
17	5/5/99	41.86	-6.07	-17.71	21.08	10.46	8.80
19	5/7/99	43.22	-5.46	-17.71	21.76	10.04	8.47
21	5/9/99	44.76	-5.07	-17.92	22.40	10.10	8.49
23	5/11/99	42.45	-4.74	-17.67	21.11	10.05	8.59
25	5/13/99	44.19	-4.97	-17.70	22.17	10.18	8.59
27	5/15/99	43.13	-4.80	-17.92	21.49	10.04	8.59
29	5/17/99	44.87	-4.45	-17.80	22.38	10.14	8.59
31	5/19/99	44.48	-3.93	-17.67	22.14	10.04	8.59
33	5/21/99	41.71	-3.80	-16.52	21.59	9.29	8.09
35	5/23/99	42.90	-3.43	-17.60	22.21	9.52	8.50
37	5/25/99	44.92	-3.53	-17.65	22.55	9.96	8.59
39	5/27/99	46.21	-2.83	-17.32	22.84	10.41	8.87
41	5/29/99	47.81	-2.45	-17.21	23.91	10.51	8.94
43	5/31/99	47.34	-2.74	-17.70	23.61	9.96	8.54
45	6/2/99	49.07	-1.86	-17.56	24.55	10.63	9.14
47	6/4/99	47.59	-1.26	-17.71	23.84	10.96	9.70
49	6/6/99	46.02	-1.61	-17.95	22.90	10.24	9.13
51	6/8/99	47.59	-1.02	-17.84	23.48	10.15	8.97
53	6/10/99	47.38	-0.57	-17.99	23.76	10.10	9.15
55	6/12/99	46.89	-0.65	-17.86	23.94	9.71	8.88
57	6/14/99	45.94	-0.62	-17.75	23.48	9.63	8.87
59	6/16/99	46.96	-0.73	-17.91	23.74	9.76	8.86
61	6/18/99	46.44	-0.97	-18.04	23.32	9.78	8.87
63	6/20/99	46.05	-0.98	-17.81	23.09	9.81	8.87
65	6/22/99	46.23	-0.80	-17.85	23.27	9.75	8.86
67	6/24/99	45.85	-0.69	-17.75	23.17	9.70	8.87
69	6/26/99	46.30	-1.28	-17.56	23.42	9.05	8.10
71	6/28/99	45.27	-2.48	-18.03	22.21	9.12	7.99
73	6/30/99	44.92	-2.16	-17.93	22.40	9.02	8.03
75	7/2/99	47.07	-1.66	-18.16	23.69	8.99	8.03
77	7/4/99	46.78	-1.36	-18.16	23.51	9.12	8.21
79	7/6/99	44.95	-1.40	-18.01	22.68	9.11	8.31
81	7/8/99	43.56	-1.59	-17.92	21.93	9.05	8.31
83	7/10/99	46.14	-0.81	-18.07	23.17	9.73	8.89
85	7/12/99	47.74	-0.18	-17.99	24.18	9.84	9.00
87	7/14/99	45.10	-0.55	-17.89	22.72	9.74	9.01
89	7/16/99	46.38	-0.47	-17.98	23.35	9.82	9.00
91	7/18/99	47.03	-0.29	-17.98	23.72	9.83	9.00
93	7/20/99	47.06	-0.21	-17.98	23.68	9.83	9.00
95	7/22/99	46.26	-0.43	-17.89	23.37	9.80	9.00
97	7/24/99	46.35	-0.87	-17.80	23.67	8.95	8.17

Table A3: Samples of the Results Using BD Method for Exchanger Case 1

Day	Re_s	h_c ($W/m^2 K$)	J_C	J_L	J_B	h_s ($W/m^2 K$)	Re_t
1	6564.26	929.62	1.02	0.86	0.93	758.44	9823.69
3	7458.59	1010.23	1.02	0.86	0.93	824.21	10679.41
5	7398.37	1004.91	1.02	0.86	0.93	819.87	10937.63
7	8030.99	1060.04	1.02	0.86	0.93	864.85	10938.33
9	7991.64	1056.66	1.02	0.86	0.93	862.09	10937.90
11	7419.74	1006.80	1.02	0.86	0.93	821.41	10179.71
13	7119.38	980.07	1.02	0.86	0.93	799.61	9825.82
15	7355.28	1001.10	1.02	0.86	0.93	816.76	9815.34
17	6555.07	928.78	1.02	0.86	0.93	757.75	8910.14
19	6294.36	904.56	1.02	0.86	0.93	738.00	8570.03
21	6327.78	907.68	1.02	0.86	0.93	740.54	8597.00
23	6299.61	905.05	1.02	0.86	0.93	738.40	8695.06
25	6380.93	912.64	1.02	0.86	0.93	744.59	8691.77
27	6294.74	904.59	1.02	0.86	0.93	738.02	8693.47
29	6354.36	910.16	1.02	0.86	0.93	742.57	8693.78
31	6293.97	904.52	1.02	0.86	0.93	737.97	8694.01
33	5825.40	860.09	1.02	0.86	0.93	701.72	8192.36
35	5964.11	873.37	1.02	0.86	0.93	712.55	8605.44
37	6242.30	899.68	1.02	0.86	0.93	734.02	8698.89
39	6527.26	926.21	1.02	0.86	0.93	755.66	8976.30
41	6585.16	931.55	1.02	0.86	0.93	760.02	9048.11
43	6243.84	899.83	1.02	0.86	0.93	734.13	8648.33
45	6660.39	938.46	1.02	0.86	0.93	765.66	9250.45
47	6871.41	957.72	1.02	0.86	0.93	781.36	9816.03
49	6416.48	915.95	1.02	0.86	0.93	747.28	9239.93
51	6360.39	910.72	1.02	0.86	0.93	743.03	9081.28
53	6332.85	908.16	1.02	0.86	0.93	740.93	9261.89
55	6084.71	884.83	1.02	0.86	0.93	721.90	8985.68
57	6038.01	880.40	1.02	0.86	0.93	718.29	8978.34
59	6116.99	887.88	1.02	0.86	0.93	724.39	8972.81
61	6126.76	888.81	1.02	0.86	0.93	725.14	8975.56
63	6146.20	890.64	1.02	0.86	0.93	726.64	8975.65
65	6111.68	887.38	1.02	0.86	0.93	723.98	8974.29
67	6078.82	884.27	1.02	0.86	0.93	721.44	8975.89
69	5669.34	845.02	1.02	0.86	0.93	689.42	8200.09
71	5715.43	849.49	1.02	0.86	0.93	693.07	8088.25
73	5655.26	843.66	1.02	0.86	0.93	688.31	8133.42
75	5636.89	841.87	1.02	0.86	0.93	686.85	8133.05
77	5718.63	849.80	1.02	0.86	0.93	693.32	8314.29
79	5708.04	848.77	1.02	0.86	0.93	692.48	8415.42
81	5673.89	845.47	1.02	0.86	0.93	689.78	8413.26
83	6100.52	886.33	1.02	0.86	0.93	723.12	9000.82
85	6168.12	892.71	1.02	0.86	0.93	728.33	9116.52
87	6103.28	886.59	1.02	0.86	0.93	723.33	9116.76
89	6152.41	891.23	1.02	0.86	0.93	727.12	9115.32
91	6159.00	891.85	1.02	0.86	0.93	727.62	9115.15
93	6159.10	891.86	1.02	0.86	0.93	727.63	9115.80
95	6142.28	890.27	1.02	0.86	0.93	726.34	9115.21
97	5608.81	839.14	1.02	0.86	0.93	684.62	8272.57

Table A3: Samples of the Results Using BD Method for Exchanger Case 1 (cont.)

Day	h_t (W/m^2 K)	R_k (W/m^2 K)	U_c (W/m^2 K)	LMTD	F	U_a (W/m^2 K)	Fouling Resistance (hr. ft². F/BTU)
1	792.16	5.01E-05	334.47	14.37	0.78	287.13	0.00280
3	846.90	5.01E-05	359.74	15.18	0.80	297.88	0.00328
5	863.24	5.01E-05	362.65	14.72	0.81	297.04	0.00346
7	863.29	5.01E-05	371.20	16.02	0.83	283.35	0.00475
9	863.26	5.01E-05	370.68	15.83	0.83	284.97	0.00461
11	815.05	5.01E-05	351.72	15.75	0.83	260.06	0.00569
13	792.30	5.01E-05	342.28	15.84	0.83	249.67	0.00616
15	791.62	5.01E-05	345.22	16.54	0.84	245.31	0.00670
17	732.66	5.01E-05	320.24	15.77	0.84	222.32	0.00782
19	710.20	5.01E-05	311.20	16.43	0.85	206.38	0.00927
21	711.99	5.01E-05	312.09	17.17	0.86	201.04	0.01006
23	718.48	5.01E-05	313.30	16.79	0.87	195.17	0.01098
25	718.26	5.01E-05	314.36	16.95	0.86	203.06	0.00991
27	718.37	5.01E-05	313.21	17.03	0.86	195.50	0.01092
29	718.39	5.01E-05	314.03	17.52	0.87	195.14	0.01102
31	718.41	5.01E-05	313.21	17.69	0.87	188.18	0.01205
33	685.05	5.01E-05	298.52	16.14	0.86	185.29	0.01163
35	712.55	5.01E-05	307.14	17.22	0.87	183.42	0.01248
37	718.73	5.01E-05	312.57	17.93	0.88	185.48	0.01245
39	737.01	5.01E-05	320.93	18.58	0.88	185.28	0.01296
41	741.72	5.01E-05	322.86	18.96	0.88	187.69	0.01267
43	715.39	5.01E-05	311.78	19.01	0.88	178.25	0.01365
45	754.96	5.01E-05	327.09	19.78	0.89	185.10	0.01333
47	791.67	5.01E-05	338.74	19.88	0.90	185.91	0.01379
49	754.27	5.01E-05	323.53	19.53	0.90	173.48	0.01519
51	743.90	5.01E-05	320.28	20.25	0.90	167.04	0.01627
53	755.71	5.01E-05	322.67	20.36	0.90	166.79	0.01646
55	737.62	5.01E-05	314.82	19.94	0.90	164.50	0.01649
57	737.14	5.01E-05	314.02	19.67	0.90	163.03	0.01676
59	736.78	5.01E-05	315.10	20.05	0.90	163.69	0.01668
61	736.96	5.01E-05	315.28	19.94	0.90	163.62	0.01671
63	736.97	5.01E-05	315.57	19.74	0.90	164.03	0.01663
65	736.88	5.01E-05	315.04	19.86	0.90	162.85	0.01686
67	736.98	5.01E-05	314.59	19.74	0.90	162.12	0.01699
69	685.57	5.01E-05	296.40	19.39	0.90	155.23	0.01743
71	678.08	5.01E-05	295.25	19.06	0.89	156.77	0.01700
73	681.10	5.01E-05	295.12	18.95	0.89	156.39	0.01708
75	681.08	5.01E-05	294.85	19.74	0.89	155.06	0.01737
77	693.19	5.01E-05	298.95	19.86	0.90	155.11	0.01762
79	699.93	5.01E-05	300.39	19.30	0.90	155.44	0.01764
81	699.79	5.01E-05	299.85	18.86	0.90	154.92	0.01773
83	738.62	5.01E-05	315.29	19.98	0.90	161.85	0.01709
85	746.20	5.01E-05	318.05	20.55	0.90	162.40	0.01712
87	746.22	5.01E-05	317.10	19.75	0.90	160.85	0.01740
89	746.13	5.01E-05	317.80	20.14	0.90	161.92	0.01721
91	746.11	5.01E-05	317.90	20.37	0.90	161.65	0.01727
93	746.16	5.01E-05	317.91	20.45	0.91	160.72	0.01748
95	746.12	5.01E-05	317.65	20.05	0.90	162.18	0.01715
97	690.41	5.01E-05	296.66	19.67	0.90	152.80	0.01803

Table A4: Samples of the Results Using FS Method for Exchanger Case 1

Day	Date	Re_s	F_b	F_s	F_t	h_s (W/m^2 K)	Re_t
1	4/19/99	3318.02	0.15	0.13	0.21	596.23	9823.69
3	4/21/99	3784.89	0.15	0.13	0.21	649.58	10679.41
5	4/23/99	3753.35	0.15	0.13	0.21	646.05	10937.63
7	4/25/99	4085.53	0.15	0.13	0.21	682.72	10938.33
9	4/27/99	4064.81	0.15	0.13	0.21	680.46	10937.90
11	4/29/99	3764.53	0.15	0.13	0.21	647.30	10179.71
13	5/1/99	3607.41	0.15	0.13	0.21	629.58	9825.82
15	5/3/99	3730.79	0.15	0.13	0.21	643.52	9815.34
17	5/5/99	3313.24	0.15	0.13	0.21	595.67	8910.14
19	5/7/99	3177.76	0.15	0.13	0.21	579.70	8570.03
21	5/9/99	3195.11	0.15	0.13	0.21	581.75	8597.00
23	5/11/99	3180.49	0.15	0.13	0.21	580.02	8695.06
25	5/13/99	3222.72	0.15	0.13	0.21	585.02	8691.77
27	5/15/99	3177.96	0.15	0.13	0.21	579.72	8693.47
29	5/17/99	3208.92	0.15	0.13	0.21	583.39	8693.78
31	5/19/99	3177.56	0.15	0.13	0.21	579.67	8694.01
33	5/21/99	2829.73	0.16	0.13	0.22	537.53	8192.36
35	5/23/99	2898.26	0.16	0.13	0.22	545.97	8605.44
37	5/25/99	3150.73	0.15	0.13	0.21	576.48	8698.89
39	5/27/99	3298.78	0.15	0.13	0.21	593.97	8976.30
41	5/29/99	3328.90	0.15	0.13	0.21	597.50	9048.11
43	5/31/99	3151.53	0.15	0.13	0.21	576.58	8648.33
45	6/2/99	3368.05	0.15	0.13	0.21	602.06	9250.45
47	6/4/99	3477.99	0.15	0.13	0.21	614.79	9816.03
49	6/6/99	3241.19	0.15	0.13	0.21	587.20	9239.93
51	6/8/99	3212.05	0.15	0.13	0.21	583.76	9081.28
53	6/10/99	3197.74	0.15	0.13	0.21	582.07	9261.89
55	6/12/99	3068.97	0.15	0.13	0.21	566.70	8985.68
57	6/14/99	3044.76	0.15	0.13	0.21	563.78	8978.34
59	6/16/99	3085.72	0.15	0.13	0.21	568.71	8972.81
61	6/18/99	3090.78	0.15	0.13	0.21	569.32	8975.56
63	6/20/99	3100.87	0.15	0.13	0.21	570.53	8975.65
65	6/22/99	3082.96	0.15	0.13	0.21	568.38	8974.29
67	6/24/99	3065.92	0.15	0.13	0.21	566.33	8975.89
69	6/26/99	2752.64	0.16	0.13	0.22	527.95	8200.09
71	6/28/99	2775.41	0.16	0.13	0.22	530.79	8088.25
73	6/30/99	2745.68	0.16	0.13	0.22	527.09	8133.42
75	7/2/99	2736.61	0.16	0.13	0.22	525.95	8133.05
77	7/4/99	2776.99	0.16	0.13	0.22	530.99	8314.29
79	7/6/99	2771.76	0.16	0.13	0.22	530.34	8415.42
81	7/8/99	2754.89	0.16	0.13	0.22	528.23	8413.26
83	7/10/99	3077.18	0.15	0.13	0.21	567.68	9000.82
85	7/12/99	3112.24	0.15	0.13	0.21	571.89	9116.52
87	7/14/99	3078.60	0.15	0.13	0.21	567.86	9116.76
89	7/16/99	3104.09	0.15	0.13	0.21	570.91	9115.32
91	7/18/99	3107.51	0.15	0.13	0.21	571.32	9115.15
93	7/20/99	3107.56	0.15	0.13	0.21	571.33	9115.80
95	7/22/99	3098.84	0.15	0.13	0.21	570.28	9115.21
97	7/24/99	2722.74	0.16	0.13	0.22	524.21	8272.57

Table A4: Samples of the Results Using FS Method for Exchanger Case 1 (cont.)

Day	R_k (W/m² K)	h_t (W/m² K)	U_c (W/m² K)	LMTD	F	U_a (W/m² K)	Fouling Resistance (hr. ft².F/BTU)
1	5.01E-05	792.16	302.55	14.37	0.78	287.13	0.00101
3	5.01E-05	846.90	326.51	15.18	0.80	297.88	0.00167
5	5.01E-05	863.24	328.69	14.72	0.81	297.04	0.00184
7	5.01E-05	863.29	337.93	16.02	0.83	283.35	0.00324
9	5.01E-05	863.26	337.38	15.83	0.83	284.97	0.00310
11	5.01E-05	815.05	319.76	15.75	0.83	260.06	0.00408
13	5.01E-05	792.30	310.94	15.84	0.83	249.67	0.00448
15	5.01E-05	791.62	314.16	16.54	0.84	245.31	0.00508
17	5.01E-05	732.66	290.83	15.77	0.84	222.32	0.00602
19	5.01E-05	710.20	282.48	16.43	0.85	206.38	0.00742
21	5.01E-05	711.99	283.33	17.17	0.86	201.04	0.00821
23	5.01E-05	718.48	284.23	16.79	0.87	195.17	0.00912
25	5.01E-05	718.26	285.38	16.95	0.86	203.06	0.00807
27	5.01E-05	718.37	284.13	17.03	0.86	195.50	0.00907
29	5.01E-05	718.39	285.01	17.52	0.87	195.14	0.00918
31	5.01E-05	718.41	284.13	17.69	0.87	188.18	0.01020
33	5.01E-05	685.05	267.25	16.14	0.86	185.29	0.00940
35	5.01E-05	712.55	274.67	17.22	0.87	183.42	0.01029
37	5.01E-05	718.73	283.42	17.93	0.88	185.48	0.01059
39	5.01E-05	737.01	291.30	18.58	0.88	185.28	0.01116
41	5.01E-05	741.72	293.09	18.96	0.88	187.69	0.01089
43	5.01E-05	715.39	282.78	19.01	0.88	178.25	0.01178
45	5.01E-05	754.96	296.84	19.78	0.89	185.10	0.01155
47	5.01E-05	791.67	307.16	19.88	0.90	185.91	0.01206
49	5.01E-05	754.27	293.05	19.53	0.90	173.48	0.01336
51	5.01E-05	743.90	290.17	20.25	0.90	167.04	0.01443
53	5.01E-05	755.71	292.04	20.36	0.90	166.79	0.01461
55	5.01E-05	737.62	284.70	19.94	0.90	164.50	0.01458
57	5.01E-05	737.14	283.87	19.67	0.90	163.03	0.01484
59	5.01E-05	736.78	285.04	20.05	0.90	163.69	0.01478
61	5.01E-05	736.96	285.23	19.94	0.90	163.62	0.01481
63	5.01E-05	736.97	285.54	19.74	0.90	164.03	0.01474
65	5.01E-05	736.88	284.98	19.86	0.90	162.85	0.01495
67	5.01E-05	736.98	284.48	19.74	0.90	162.12	0.01507
69	5.01E-05	685.57	264.96	19.39	0.90	155.23	0.01516
71	5.01E-05	678.08	264.22	19.06	0.89	156.77	0.01474
73	5.01E-05	681.10	263.88	18.95	0.89	156.39	0.01480
75	5.01E-05	681.08	263.59	19.74	0.89	155.06	0.01509
77	5.01E-05	693.19	267.18	19.86	0.90	155.11	0.01537
79	5.01E-05	699.93	268.30	19.30	0.90	155.44	0.01537
81	5.01E-05	699.79	267.73	18.86	0.90	154.92	0.01545
83	5.01E-05	738.62	285.14	19.98	0.90	161.85	0.01518
85	5.01E-05	746.20	287.65	20.55	0.90	162.40	0.01523
87	5.01E-05	746.22	286.63	19.75	0.90	160.85	0.01550
89	5.01E-05	746.13	287.39	20.14	0.90	161.92	0.01532
91	5.01E-05	746.11	287.49	20.37	0.90	161.65	0.01538
93	5.01E-05	746.16	287.50	20.45	0.91	160.72	0.01559
95	5.01E-05	746.12	287.23	20.05	0.90	162.18	0.01525
97	5.01E-05	690.41	264.93	19.67	0.90	152.80	0.01574

Table A5: Statistical Functions for the Fouling Cycles for Exchanger Case 1

Data # (fouling Cycle)	Time (days)	Cumulative Distribution Function F(t)	Reliability R(t)	Hazard Function H(t)
1	59	0.111	0.889	0.118
2	61	0.222	0.778	0.251
3	65	0.333	0.667	0.405
4	87	0.444	0.556	0.588
5	97	0.556	0.444	0.811
6	97	0.667	0.333	1.099
7	99	0.778	0.222	1.504
8	127	0.889	0.111	2.197

Table A6: Statistical Distributions for the Fouling Cycles for Exchanger Case 1

Data # (fouling Cycle)	Time	F(t)	Exponential Distribution		Weibull Distribution		Normal Distribution		Lognormal Distribution	
			X	Y	X	X	Y	Y	X	Y
1	59	0.11	59	0.12	4.08	4.08	-1.22	-2.14	59	-1.22
2	61	0.22	61	0.25	4.11	4.11	-0.76	-1.38	61	-0.76
3	65	0.33	65	0.41	4.17	4.17	-0.43	-0.90	65	-0.43
4	87	0.44	87	0.59	4.47	4.47	-0.14	-0.53	87	-0.14
5	97	0.55	97	0.81	4.57	4.57	0.14	-0.21	97	0.14
6	97	0.66	97	1.10	4.57	4.57	0.43	0.09	97	0.43
7	99	0.77	99	1.50	4.60	4.60	0.76	0.41	99	0.76
8	127	0.88	127	2.20	4.84	4.84	1.22	0.78	127	1.22

Table A7: Operation Data for the Modified Heat Exchanger Case 1 (HBHE)

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	3/20/01	40.67	-6.83	-17.25	24.48	15.10	14.67
2	3/21/01	41.33	-6.18	-16.54	25.06	15.19	14.71
3	3/22/01	41.04	-6.93	-17.27	24.55	14.95	14.42
4	3/23/01	39.83	-6.97	-17.24	23.84	14.85	14.42
5	3/24/01	41.81	-6.84	-17.26	24.90	15.04	14.42
6	3/25/01	42.28	-6.91	-17.27	24.97	15.18	14.42
7	3/26/01	40.42	-7.11	-17.24	23.75	15.11	14.42
8	3/27/01	40.24	-7.13	-17.21	23.57	15.15	14.42
9	3/28/01	41.33	-6.93	-17.21	24.32	15.15	14.42
10	3/29/01	42.54	-6.78	-17.25	25.10	15.18	14.42
11	3/30/01	44.29	-6.66	-17.31	25.98	15.36	14.44
12	3/31/01	43.63	-6.71	-17.31	25.59	15.30	14.42
13	4/1/01	44.84	-6.66	-17.54	26.29	15.63	14.71
14	4/2/01	46.24	-5.50	-16.36	27.44	15.71	14.71
15	4/3/01	46.47	-6.33	-17.45	27.14	16.41	15.32
16	4/4/01	46.67	-6.28	-17.46	27.33	16.24	15.19
17	4/4/01	47.19	-6.61	-17.62	27.13	16.79	15.45
18	4/6/01	44.61	-6.74	-17.55	25.68	16.74	15.59
19	4/7/01	46.70	-6.47	-17.66	27.19	16.32	15.22
20	4/8/01	46.69	-7.39	-18.45	26.53	16.63	15.30
21	4/9/01	45.92	-7.96	-19.20	26.18	15.79	14.71
22	4/10/01	44.57	-6.72	-17.51	25.73	16.14	15.05
23	4/11/01	46.42	-6.30	-17.53	27.12	16.11	15.09
24	4/12/01	45.10	-6.66	-17.33	25.94	15.91	14.71
25	4/13/01	46.64	-6.30	-17.35	26.90	16.65	15.39
26	4/14/01	46.03	-6.59	-17.40	26.19	16.86	15.45
27	4/15/01	47.12	-6.60	-17.68	26.74	16.91	15.46
28	4/16/01	49.10	-6.19	-17.48	28.03	16.97	15.45
29	4/17/01	51.86	-5.26	-16.98	30.12	16.83	15.35

Table A8: Results for Exchanger Case 1 to Develop J_H Factor

Day	Re_s	h_c (W/m^2 K)	J_C	J_L	J_B	h_s (W/m^2 K)	Re_t
1	9462.95	1179.54	1.02	0.86	0.93	962.34	14854.20
2	9520.41	1184.20	1.02	0.86	0.93	966.14	14893.85
3	9371.01	1172.07	1.02	0.86	0.93	956.25	14596.24
4	9307.78	1166.91	1.02	0.86	0.93	952.04	14596.61
5	9428.60	1176.75	1.02	0.86	0.93	960.07	14597.47
6	9515.61	1183.81	1.02	0.86	0.93	965.83	14597.92
7	9470.85	1180.18	1.02	0.86	0.93	962.86	14596.67
8	9492.67	1181.95	1.02	0.86	0.93	964.31	14597.35
9	9494.47	1182.10	1.02	0.86	0.93	964.43	14597.46
10	9514.56	1183.72	1.02	0.86	0.93	965.76	14597.60
11	9630.02	1193.06	1.02	0.86	0.93	973.37	14622.27
12	9587.05	1189.59	1.02	0.86	0.93	970.54	14597.24
13	9795.04	1206.33	1.02	0.86	0.93	984.19	14895.45
14	9847.97	1210.57	1.02	0.86	0.93	987.65	14895.61
15	10283.94	1245.19	1.02	0.86	0.93	1015.90	15513.58
16	10177.13	1236.76	1.02	0.86	0.93	1009.02	15379.84
17	10526.24	1264.21	1.02	0.86	0.93	1031.42	15641.54
18	10494.95	1261.76	1.02	0.86	0.93	1029.42	15787.57
19	10226.21	1240.63	1.02	0.86	0.93	1012.19	15410.51
20	10425.22	1256.30	1.02	0.86	0.93	1024.97	15492.21
21	9898.25	1214.59	1.02	0.86	0.93	990.93	14895.89
22	10114.93	1231.83	1.02	0.86	0.93	1005.00	15235.41
23	10094.92	1230.24	1.02	0.86	0.93	1003.71	15275.55
24	9969.71	1220.29	1.02	0.86	0.93	995.59	14894.73
25	10433.17	1256.92	1.02	0.86	0.93	1025.48	15580.81
26	10564.80	1267.22	1.02	0.86	0.93	1033.88	15640.13
27	10597.72	1269.79	1.02	0.86	0.93	1035.98	15655.94
28	10634.29	1272.64	1.02	0.86	0.93	1038.30	15638.59
29	10545.90	1265.75	1.02	0.86	0.93	1032.67	15537.22

Table A8: Results for Exchanger Case 1 to Develop J_H Factor (cont.)

Day	h_t (W/m^2 K)	R_k (W/m^2 K)	U_a (W/m^2 K)	$\dot{Q}$ (W)	LMTD (°C)	F	J_H
1	1102.74	5.0E-05	496.68	1292831	13.09	0.73	1.30
2	1105.10	5.0E-05	497.55	1296433	13.09	0.73	1.30
3	1087.40	5.0E-05	489.25	1282858	13.18	0.73	1.29
4	1087.42	5.0E-05	486.15	1251692	12.92	0.73	1.27
5	1087.47	5.0E-05	485.14	1301093	13.40	0.73	1.25
6	1087.50	5.0E-05	483.73	1315694	13.54	0.73	1.24
7	1087.42	5.0E-05	481.09	1270863	13.13	0.73	1.23
8	1087.46	5.0E-05	480.41	1267058	13.10	0.74	1.22
9	1087.47	5.0E-05	478.66	1290692	13.36	0.74	1.21
10	1087.48	5.0E-05	477.94	1318935	13.66	0.74	1.20
11	1088.95	5.0E-05	474.71	1364771	14.14	0.74	1.17
12	1087.45	5.0E-05	472.89	1346388	14.00	0.74	1.17
13	1105.19	5.0E-05	478.51	1405580	14.38	0.75	1.16
14	1105.20	5.0E-05	475.80	1412115	14.47	0.75	1.14
15	1141.73	5.0E-05	490.03	1500858	14.85	0.75	1.14
16	1133.85	5.0E-05	485.96	1492146	14.89	0.75	1.13
17	1149.26	5.0E-05	493.51	1541735	15.08	0.76	1.13
18	1157.84	5.0E-05	494.48	1485264	14.49	0.76	1.13
19	1135.66	5.0E-05	485.56	1501239	14.97	0.75	1.12
20	1140.47	5.0E-05	488.94	1534901	15.16	0.76	1.12
21	1105.22	5.0E-05	473.25	1470513	15.09	0.75	1.12
22	1125.33	5.0E-05	479.42	1431457	14.44	0.76	1.11
23	1127.70	5.0E-05	478.30	1475498	14.90	0.76	1.10
24	1105.15	5.0E-05	471.25	1412335	14.51	0.75	1.10
25	1145.69	5.0E-05	484.32	1511327	14.98	0.76	1.09
26	1149.18	5.0E-05	485.55	1507689	14.87	0.76	1.08
27	1150.11	5.0E-05	481.42	1540923	15.26	0.77	1.06
28	1149.09	5.0E-05	481.44	1584104	15.68	0.77	1.06
29	1143.12	5.0E-05	477.56	1626022	16.22	0.77	1.05

Table A9: Sample of the collected data for Exchanger Case 1 after HBHE Installation

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	3/20/01	41.33	-6.19	-16.54	25.06	15.15	14.67
2	3/21/01	40.67	-6.83	-17.25	24.48	15.10	14.67
3	3/22/01	41.05	-6.93	-17.27	24.55	14.95	14.42
4	3/23/01	39.83	-6.97	-17.24	23.84	14.85	14.42
5	3/24/01	41.81	-6.84	-17.26	24.90	15.04	14.42
6	3/25/01	42.28	-6.91	-17.27	24.96	15.18	14.42
7	3/26/01	40.41	-7.11	-17.24	23.75	15.11	14.42
8	3/27/01	40.25	-7.13	-17.21	23.57	15.14	14.42
9	3/28/01	41.33	-6.93	-17.21	24.32	15.15	14.42
10	3/29/01	42.54	-6.78	-17.25	25.10	15.18	14.42
11	3/30/01	43.63	-6.71	-17.31	25.59	15.29	14.42
12	3/31/01	44.29	-6.66	-17.31	25.98	15.37	14.44
13	4/1/01	44.85	-6.66	-17.54	26.29	15.63	14.71
14	4/2/01	46.25	-5.50	-16.36	27.44	15.71	14.71
15	4/3/01	45.92	-7.96	-19.20	26.18	15.79	14.71
16	4/4/01	46.67	-6.28	-17.46	27.33	16.24	15.19
17	4/5/01	46.47	-6.33	-17.45	27.14	16.41	15.32
18	4/6/01	46.70	-6.47	-17.66	27.19	16.32	15.22
19	4/7/01	46.69	-7.39	-18.45	26.53	16.63	15.30
20	4/8/01	47.19	-6.61	-17.62	27.13	16.79	15.45
21	4/9/01	44.61	-6.74	-17.55	25.69	16.74	15.59
22	4/10/01	44.56	-6.72	-17.51	25.73	16.14	15.05
23	4/11/01	45.10	-6.65	-17.33	25.95	15.90	14.71
29	4/12/01	46.42	-6.30	-17.53	27.12	16.11	15.09
30	4/13/01	46.64	-6.30	-17.35	26.90	16.65	15.39
31	4/14/01	46.03	-6.59	-17.40	26.20	16.86	15.45
33	4/15/01	46.98	-6.07	-17.62	27.56	16.02	15.09
35	4/16/01	47.12	-6.60	-17.68	26.75	16.91	15.46
36	4/17/01	49.10	-6.19	-17.48	28.03	16.97	15.45
37	4/18/01	51.86	-5.26	-16.98	30.12	16.82	15.35
38	4/19/01	47.40	-6.60	-18.46	26.93	16.14	15.01
39	4/20/01	45.61	-6.00	-17.60	26.27	15.96	15.01
40	4/21/01	46.34	-6.15	-17.92	26.63	16.04	15.06
41	4/22/01	47.08	-5.73	-17.54	27.44	14.86	14.00
50	4/23/01	49.27	-5.55	-17.45	28.21	15.97	14.71
51	4/24/01	46.05	-5.67	-17.45	26.54	15.64	14.71
52	4/25/01	46.72	-5.63	-17.48	26.90	15.69	14.71
53	4/26/01	47.37	-5.59	-17.49	27.28	15.73	14.71
54	4/27/01	48.72	-5.16	-17.44	28.27	15.68	14.71
55	4/28/01	48.38	-4.97	-17.36	28.25	15.56	14.71
56	4/29/01	50.62	-4.85	-16.99	29.14	15.99	14.71
57	4/30/01	54.52	-4.86	-17.57	31.25	16.17	14.71
58	5/1/01	50.06	-5.21	-17.52	28.76	15.88	14.71
59	5/2/01	49.48	-5.28	-17.52	28.36	15.87	14.71
60	5/3/01	48.75	-5.50	-17.50	27.74	15.95	14.71
61	5/4/01	47.97	-5.50	-17.54	27.38	15.83	14.71
62	5/5/01	51.42	-4.93	-17.56	29.62	15.98	14.80
63	5/6/01	49.58	-5.06	-17.59	28.57	16.08	15.03
64	5/7/01	49.90	-4.92	-17.56	28.79	16.21	15.15

Table A10: Sample of Results for Exchanger Case 1 after HBHE Installation

Day	U_a (W/m^2 K)	h_t (W/m^2 K)	h_c (W/m^2 K)	J_H	h_s (W/m^2 K)	U_c (W/m^2 K)	Fouling Resistance (hr. ft^2 F/BTU)
1	496.38	1102.74	1182.15	1.24	1191.60	486.88	0.0000
2	496.71	1102.74	1179.56	1.25	1199.46	488.19	0.0000
3	489.37	1087.40	1172.06	1.27	1217.88	487.28	0.0000
4	486.02	1087.42	1166.88	1.29	1224.48	488.34	0.0001
5	485.19	1087.47	1176.81	1.26	1207.11	485.56	0.0000
6	483.74	1087.50	1183.88	1.23	1186.11	482.14	0.0000
7	481.09	1087.42	1180.17	1.24	1197.64	484.01	0.0001
8	480.33	1087.46	1181.91	1.24	1192.36	483.16	0.0001
9	478.71	1087.47	1182.08	1.24	1191.81	483.07	0.0001
10	478.00	1087.48	1183.77	1.23	1186.45	482.19	0.0001
11	472.81	1087.45	1189.53	1.20	1167.68	479.05	0.0002
12	474.73	1088.95	1193.12	1.19	1156.21	477.48	0.0001
13	478.35	1105.19	1206.31	1.14	1122.36	475.49	0.0000
14	475.87	1105.20	1210.66	1.13	1115.66	474.29	0.0000
15	473.14	1105.22	1214.56	1.12	1111.92	473.62	0.0000
16	486.05	1133.85	1236.73	1.12	1127.58	483.19	0.0000
17	490.01	1141.73	1245.26	1.12	1141.53	487.58	0.0000
18	485.43	1135.66	1240.63	1.12	1134.16	484.82	0.0000
19	489.02	1140.47	1256.35	1.12	1148.66	488.58	0.0000
20	493.41	1149.26	1264.19	1.10	1133.71	487.90	0.0000
21	494.65	1157.84	1261.78	1.11	1140.85	491.21	0.0000
22	479.43	1125.33	1231.86	1.11	1120.01	479.81	0.0000
23	471.20	1105.15	1220.21	1.12	1110.37	473.32	0.0001
29	478.18	1127.70	1230.24	1.11	1117.83	479.96	0.0000
30	484.17	1145.69	1256.92	1.12	1148.30	489.75	0.0001
31	485.66	1149.18	1267.22	1.08	1120.74	485.46	0.0000
33	471.22	1127.84	1225.99	1.11	1113.26	479.15	0.0002
35	481.35	1150.11	1269.78	1.07	1105.73	482.83	0.0000
36	481.43	1149.09	1272.61	1.04	1084.32	478.48	0.0000
37	477.49	1143.12	1265.68	1.09	1127.92	485.41	0.0002
38	447.10	1122.74	1232.03	1.11	1120.26	479.25	0.0009
39	445.95	1122.78	1222.91	1.11	1111.19	477.59	0.0008
40	446.93	1125.93	1227.01	1.11	1114.18	478.88	0.0008
41	420.91	1061.89	1167.37	1.29	1224.14	481.60	0.0017
50	435.55	1105.18	1223.42	1.11	1111.45	473.52	0.0010
51	429.74	1105.13	1206.75	1.14	1121.57	475.34	0.0013
52	430.02	1105.19	1209.39	1.13	1117.35	474.59	0.0012
53	431.24	1105.16	1211.54	1.13	1114.63	474.09	0.0012
54	427.31	1105.18	1208.90	1.13	1113.06	474.72	0.0013
55	424.22	1105.20	1202.99	1.15	1129.15	476.71	0.0015
56	429.55	1105.24	1224.61	1.11	1112.19	473.67	0.0012
57	430.19	1105.16	1233.66	1.12	1122.65	475.54	0.0013
58	426.74	1105.26	1219.21	1.12	1110.32	473.33	0.0013
59	425.45	1105.10	1218.66	1.12	1110.35	473.30	0.0014
60	426.49	1105.21	1222.52	1.11	1111.01	473.45	0.0013
61	423.85	1105.17	1216.54	1.12	1110.86	473.41	0.0014
62	425.82	1110.58	1224.18	1.11	1111.90	474.87	0.0014
63	428.30	1123.95	1228.88	1.11	1116.17	478.78	0.0014
64	429.91	1131.62	1235.15	1.12	1125.00	482.19	0.0014

Appendix B
Data and Sample Results for Exchanger Case 2

Table B1: Construction, Process, and Thermal Data for Exchanger Case 2

Shell internal diameter D_s (m)	0.498475	
Number of tubes N_T	3876	
Tube outside diameter D_o (m)	0.01905	
Tube inside diameter D_i (m)	0.0148336	
Tube pitch P_T (m)	0.0254	
Baffle Spacing L_B (m)	0.381	
Shell Length L_S (m)	3	
Tube to baffle diametral clearance Δ_{tb} (m)	0.0008	
Shell to baffle diametral clerance Δ_{sb} (m)	0.005	
Bundle to shell diameteral clerance Δ_b (m)	0.035	
Number of sealing strips per cross flow Nss/Nc	0.20	
Thickness of baffles t_b (m)	0.005	
Number of tube-side passes n	1	
Baffle cut	25%	
L_c (m)	0.124	
	Shell Side Liquid/vapor (Condensate/steam)	**Tube Side (Crude)**
Density (kg/m^3)	809.96/2.768	800
Thermal conductivity (W/(m.K))	0.68508/0.0379	0.128
Specific heat capacity (J/(Kg. K))	4817/2537	2076.7
Viscosity (N.S/m^2)	0.00018/0.00001884	0.002
Prandtle Number Pr	1.26563/1.26511	32.448

Table B2: Samples of the Collected Data for Exchanger Case 2

Day	Date	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	$\dot{m}_s$ (kg/s)	$\dot{m}_t$ (kg/s)
1	10/15/99	156.14	74.62	55.18	75.07	12.18	745.30
2	10/16/99	157.59	75.99	55.72	74.67	12.73	817.91
3	10/17/99	157.21	78.16	55.41	74.79	12.47	784.17
4	10/18/99	157.25	78.82	56.54	75.72	12.87	815.74
5	10/19/99	157.33	77.97	58.04	75.49	12.84	892.37
6	10/20/99	157.36	77.41	58.15	75.80	12.84	882.30
7	10/21/99	157.29	77.08	56.77	75.06	12.39	823.73
8	10/22/99	157.33	75.23	58.02	75.10	12.47	885.58
9	10/23/99	157.38	73.79	57.15	75.04	12.33	837.15
10	10/24/99	157.40	73.58	57.57	74.43	12.83	924.17
11	10/25/99	157.35	74.63	58.36	74.63	12.57	936.58
12	10/26/99	157.26	74.31	57.15	74.06	12.24	879.57
13	10/27/99	157.49	79.55	58.83	74.77	13.00	987.50
14	10/28/99	157.34	77.25	59.74	74.65	12.51	1015.15
15	10/29/99	157.41	79.29	59.98	75.55	12.09	938.84
16	10/30/99	157.41	79.65	59.65	73.97	12.90	1090.18
17	10/31/99	157.37	79.10	57.55	73.88	12.06	896.15
18	11/1/99	157.29	79.41	57.40	72.60	12.18	972.74
19	11/2/99	156.75	79.51	58.87	72.31	12.95	1166.85
20	11/3/99	157.38	80.63	58.38	72.14	12.20	1075.15
21	11/4/99	157.27	80.97	58.58	72.49	12.73	1108.58
22	11/5/99	157.44	81.41	58.67	72.29	12.46	1107.65
23	11/6/99	157.54	81.08	58.35	72.03	12.78	1132.41
24	11/7/99	157.43	81.23	59.35	72.45	12.25	1131.26
25	11/8/99	156.79	82.79	59.33	72.24	12.36	1159.28
26	11/9/99	157.49	82.81	59.35	72.22	12.11	1139.36
27	11/10/99	157.38	82.84	59.12	71.16	12.76	1283.13
28	11/11/99	157.40	82.89	59.86	71.16	12.72	1360.78

Table B3: Samples of the Thermal Results for Exchanger Case 2

Day	$LMTD_v$ (^oC)	$LMTD_{pc}$ (^oC)	$LMTD_l$ (^oC)	$LMTD_{wtd}$ (^oC)	$\%\dot{Q}_v$	$\%\dot{Q}_{pc}$	$\%\dot{Q}_l$
1	79.02	85.03	47.18	75.21	0.39	83.37	16.24
2	80.14	85.08	47.81	75.55	0.52	83.34	16.14
3	79.83	85.12	50.04	76.42	0.48	83.32	16.19
4	78.92	84.12	49.30	75.54	0.49	83.48	16.03
5	79.19	83.69	46.87	74.42	0.50	83.68	15.82
6	78.90	83.46	46.22	74.01	0.50	83.70	15.80
7	79.60	84.44	47.55	75.09	0.49	83.51	16.00
8	79.58	83.93	44.40	73.55	0.50	83.68	15.82
9	79.67	84.30	44.07	73.57	0.50	83.55	15.94
10	80.28	84.50	43.38	73.43	0.50	83.61	15.88
11	80.05	84.08	43.43	73.25	0.50	83.73	15.77
12	80.59	84.89	44.61	74.19	0.49	83.56	15.95
13	79.98	83.81	47.39	74.77	0.51	83.78	15.71
14	80.03	83.53	44.26	73.37	0.50	83.92	15.58
15	79.16	82.89	45.80	73.61	0.51	83.95	15.54
16	80.74	83.96	46.57	74.61	0.50	83.91	15.59
17	80.81	84.84	48.49	75.80	0.50	83.61	15.89
18	82.06	85.67	49.00	76.54	0.49	83.60	15.91
19	82.08	85.26	47.43	75.76	0.45	83.84	15.71
20	82.56	85.56	48.96	76.52	0.50	83.73	15.77
21	82.15	85.27	49.00	76.36	0.49	83.76	15.74
22	82.43	85.36	49.29	76.53	0.51	83.76	15.73
23	82.75	85.64	49.38	76.74	0.52	83.71	15.77
24	82.27	84.99	48.37	75.98	0.51	83.86	15.63
25	82.16	85.12	49.71	76.57	0.45	83.91	15.64
26	82.53	85.13	49.70	76.58	0.51	83.86	15.63
27	83.53	85.85	50.04	77.19	0.50	83.83	15.67
28	83.54	85.55	49.26	76.75	0.50	83.93	15.56

Table B3: Samples of the Thermal Results for Exchanger Case 2 (cont.)

Day	A_v (m^2)	A_{pc} (m^2)	A_l (m^2)	L_{pc} (m)	U_c (W/m^2 K)	U_a (W/m^2 K)	Fouling Resistance (hr. ft^2. F/BTU)
1	2.09	513.56	180.53	2.21	652.56	588.08	0.000955
2	2.67	518.82	174.70	2.24	693.31	612.00	0.001089
3	2.53	525.17	168.49	2.26	675.29	593.14	0.001165
4	2.58	523.75	169.86	2.26	691.46	618.00	0.000977
5	2.45	526.58	167.15	2.27	734.18	624.28	0.001363
6	2.49	523.36	170.33	2.26	728.41	627.53	0.001254
7	2.47	524.47	169.24	2.26	698.06	598.38	0.001356
8	2.38	520.86	172.94	2.25	732.12	613.47	0.001501
9	2.47	514.25	179.47	2.22	705.36	607.12	0.001304
10	2.39	515.73	178.06	2.22	751.52	632.90	0.001417
11	2.31	521.69	172.19	2.25	759.69	620.80	0.001673
12	2.30	522.23	171.66	2.25	730.59	597.87	0.001726
13	2.36	536.88	156.95	2.31	784.27	628.16	0.0018
14	2.17	535.05	158.96	2.31	801.28	615.12	0.002146
15	2.27	538.75	155.16	2.32	762.73	592.57	0.002139
16	2.14	544.50	149.54	2.35	836.41	624.01	0.002312
17	2.27	540.13	153.78	2.33	740.45	576.21	0.002187
18	2.11	546.85	147.22	2.36	780.96	576.34	0.002583
19	1.79	550.83	143.57	2.37	873.10	617.42	0.002695
20	1.99	555.42	138.77	2.39	832.13	576.64	0.003025
21	2.01	553.20	140.97	2.38	845.77	602.48	0.002713
22	2.02	556.49	137.68	2.40	846.32	588.41	0.002943
23	2.07	554.87	139.25	2.39	856.85	602.45	0.0028
24	1.96	558.31	135.91	2.41	858.68	581.92	0.003147
25	1.73	562.75	131.70	2.43	870.31	582.68	0.003223
26	1.95	563.62	130.62	2.43	862.09	571.11	0.003358
27	1.83	566.38	127.98	2.44	924.16	596.95	0.00337
28	1.77	569.33	125.09	2.45	957.48	597.80	0.00357

Table B4: Statistical Functions for the Fouling Cycles for Exchanger Case 2

Data # (fouling Cycle)	Time (days)	Cumulative Distribution Function F(t)	Reliability R(t)	Hazard Function H(t)
1	24	0.083	0.917	0.087
2	24	0.167	0.833	0.182
3	26	0.250	0.750	0.288
4	27	0.333	0.667	0.405
5	28	0.417	0.583	0.539
6	29	0.500	0.500	0.693
7	32	0.583	0.417	0.875
8	36	0.667	0.333	1.099
9	37	0.750	0.250	1.386
10	49	0.833	0.167	1.792
11	59	0.917	0.083	2.485

Table B5: Statistical Distributions for the Fouling Cycles for Exchanger Case 2

Data # (fouling Cycle)	Time	F(t)	Exponential Distribution		Weibull Distribution		Normal Distribution		Lognormal Distribution	
			X	Y	X	X	Y	Y	X	Y
1	24	0.08	24	0.09	3.18	3.18	-1.38	-2.44	24	-1.38
2	24	0.17	24	0.18	3.18	3.18	-0.97	-1.70	24	-0.97
3	26	0.25	26	0.29	3.26	3.26	-0.67	-1.25	26	-0.67
4	27	0.33	27	0.41	3.30	3.30	-0.43	-0.90	27	-0.43
5	28	0.42	28	0.54	3.33	3.33	-0.21	-0.62	28	-0.21
6	29	0.50	29	0.69	3.37	3.37	0.00	-0.37	29	0.00
7	32	0.58	32	0.88	3.47	3.47	0.21	-0.13	32	0.21
8	36	0.67	36	1.10	3.58	3.58	0.43	0.09	36	0.43
9	37	0.75	37	1.39	3.61	3.61	0.67	0.33	37	0.67
10	49	0.83	49	1.79	3.89	3.89	0.97	0.58	49	0.97
11	59	0.92	59	2.48	4.08	4.08	1.38	0.91	59	1.38

Table B6: Data and Results for Hydraulic Analysis of SCHE Test Unit

Day	Total Flow (m^3/hr)	Control Flow (m^3/hr)	Porosity e	Pressure Drop (with Particles) Pa	Pressure Drop (W/O Particle) Pa	Velocity (m/s)	Re_{sc}
1	9.0	0.45	0.950	2000	288.3	1.06	11716.8
2	9.0	0.60	0.935	3500	288.3	1.06	7605.9
3	9.0	0.70	0.925	4000	288.3	1.06	7524.6
4	9.0	0.80	0.920	4700	288.3	1.06	7483.9
5	9.0	0.90	0.915	4900	288.3	1.06	7443.2
6	8.0	0.55	0.930	3500	235.7	0.94	6724.7
7	8.0	0.60	0.920	4700	235.7	0.94	6652.4
8	8.0	0.70	0.913	5000	235.7	0.94	6598.1
9	8.0	0.80	0.900	5600	235.7	0.94	6507.7
10	8.0	0.90	0.885	6200	235.7	0.94	6399.3
11	6.0	0.50	0.930	3800	144.6	0.71	5043.5
12	6.0	0.60	0.900	5600	144.6	0.71	4880.8
13	6.0	0.70	0.885	6200	144.6	0.71	4799.5
14	6.0	0.80	0.870	7200	144.6	0.71	4718.1
15	6.0	0.90	0.840	7800	144.6	0.71	4555.4
16	6.0	1.00	0.850	8200	144.6	0.71	4609.6
17	6.0	1.10	0.840	8700	144.6	0.71	4555.4
18	6.0	1.50	0.845	9200	144.6	0.71	4582.5
19	6.0	2.00	0.845	8800	144.6	0.71	4582.5
20	6.0	2.50	0.845	8700	144.6	0.71	4582.5
21	6.0	2.80	0.845	8600	144.6	0.71	4582.5
22	4.5	0.55	0.895	5500	89.0	0.53	3640.3
23	4.5	0.60	0.875	6800	89.0	0.53	3558.9
24	4.5	0.70	0.865	7400	89.0	0.53	3518.2
25	4.5	0.80	0.835	9500	89.0	0.53	3396.2
26	4.5	0.90	0.825	10000	89.0	0.53	3355.6
27	4.5	1.00	0.820	10200	89.0	0.53	3335.2
28	4.5	1.10	0.825	10000	89.0	0.53	3355.6
29	4.5	1.50	0.835	9500	89.0	0.53	3396.2

Table B6: Data and Results for Hydraulic Analysis of SCHE Test Unit (Cont.)

Day	Friction Factor f	Pressure drop Factor K_{sc}	Ln (K_{sc})	Ln (Re_{sc} e^4)	Calculated Pressure Drop (with Particles) Pa
1	0.00753	5.34	1.675	9.164	2232.5
2	0.00851	12.73	2.544	8.668	3909.3
3	0.00854	14.66	2.685	8.614	4329.6
4	0.00855	17.30	2.851	8.587	4558.3
5	0.00856	18.10	2.896	8.560	4800.4
6	0.00882	15.63	2.749	8.523	4111.1
7	0.00885	21.14	3.051	8.469	4555.8
8	0.00887	22.62	3.119	8.428	4924.3
9	0.00891	25.59	3.242	8.359	5614.0
10	0.00896	28.66	3.356	8.275	6586.2
11	0.00963	27.64	3.319	8.236	4115.9
12	0.00973	41.67	3.730	8.072	5623.0
13	0.00978	46.67	3.843	7.988	6598.2
14	0.00983	54.83	4.004	7.902	7764.0
15	0.00994	60.85	4.108	7.727	10842.8
16	0.00990	63.45	4.150	7.786	9687.7
17	0.00994	67.87	4.218	7.727	10842.8
18	0.00992	71.48	4.269	7.756	10247.3
19	0.00992	68.37	4.225	7.756	10247.3
20	0.00992	67.60	4.214	7.756	10247.3
21	0.00992	66.82	4.202	7.756	10247.3
22	0.01068	66.62	4.199	7.756	5963.1
23	0.01076	83.63	4.426	7.643	7396.4
24	0.01080	91.72	4.519	7.586	8252.9
25	0.01093	120.56	4.792	7.409	11554.5
26	0.01097	127.94	4.852	7.349	12961.1
27	0.01099	131.03	4.875	7.318	13734.7
28	0.01097	127.94	4.852	7.349	12961.1
29	0.01093	120.56	4.792	7.409	11554.5

Table B7: Sample of the collected Data for SCHE Test Unit

Day	Date	Total Flow (m^3/hr)	Control Flow (m^3/hr)	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)
1	6/20/00	5.30	1.07	100.0	60.7	55.0	64.3
7	6/26/00	5.20	1.14	100.0	60.7	57.5	65.2
11	6/30/00	5.00	0.82	100.0	61.8	56.8	66.0
15	7/4/00	5.30	0.90	100.0	61.3	56.7	64.9
21	7/10/00	4.80	0.70	100.0	63.0	53.7	65.0
26	7/15/00	4.90	0.55	100.0	59.8	54.5	64.1
31	7/20/00	4.80	0.98	101.0	63.8	53.0	66.1
37	7/26/00	4.30	0.58	99.0	61.2	55.0	66.3
41	7/30/00	4.60	0.78	100.0	60.0	55.6	65.2
47	8/5/00	5.30	0.90	99.0	61.7	55.7	64.6
57	8/15/00	5.20	0.93	101.0	60.0	56.3	64.2
71	8/29/00	5.10	0.79	100.0	60.5	57.0	64.9
75	9/2/00	4.70	0.52	99.0	60.0	55.0	64.2
103	9/30/00	4.90	0.76	100.0	60.4	53.6	64.0
108	10/5/00	4.40	0.68	101.0	62.1	52.1	65.5
113	10/10/00	5.10	0.70	100.0	61.7	54.0	64.5
118	10/15/00	5.10	0.92	99.0	61.4	53.9	64.1
128	10/25/00	4.90	0.67	100.0	64.0	52.5	65.0
138	11/4/00	4.80	0.66	101.0	62.8	53.1	65.1

Table B8: Sample of the Thermal Results for SCHE Test Unit

Day	Porosity e	$\dot{Q}$ (W)	LMTD (°C)	U_a (W/m^2 K)	Vt (m/s)	Re_{sc}	Re_t
1	0.850	22746.8	16.4	977.80	0.632	32961.5	13762.3
7	0.846	18478.0	13.2	980.88	0.620	32791.1	13502.6
11	0.862	21228.5	15.1	986.32	0.596	29587.7	12983.3
15	0.859	20056.3	15.0	939.29	0.632	31765.3	13762.3
21	0.872	25031.2	19.4	907.33	0.572	27321.2	12464.0
26	0.896	21708.4	16.0	953.94	0.584	24971.5	12723.6
31	0.847	29018.4	20.5	992.71	0.572	30182.2	12464.0
37	0.882	22423.7	15.9	989.01	0.513	23402.6	11165.6
41	0.860	20379.3	14.7	974.45	0.548	27487.1	11944.6
47	0.859	21768.4	16.3	940.85	0.632	31765.3	13762.3
57	0.856	18958.0	14.4	924.80	0.620	31572.0	13502.6
71	0.867	18593.4	13.7	953.49	0.608	29660.0	13243.0
75	0.899	19954.8	15.4	913.20	0.560	23686.2	12204.3
103	0.867	23517.5	17.5	943.47	0.584	28488.1	12723.6
108	0.869	27209.4	20.1	950.23	0.524	25333.4	11425.3
113	0.876	24712.7	18.2	954.95	0.608	28504.0	13243.0
118	0.856	24006.7	17.8	946.92	0.608	31021.4	13243.0
128	0.877	28266.2	21.1	940.99	0.584	27243.9	12723.6
138	0.878	26559.6	20.0	932.82	0.572	26652.5	12464.0

Table B8: Sample of the Thermal Results for SCHE Test Unit (cont.)

Day	h_t Conventional Heat Exchanger (W/m^2 K)	$V_{1,s}$ (m/s)	ht SCHE (W/m^2 K)	Re_{dp}	Ar	U_c (W/m^2 K)	Fouling Resistance (hr. ft^2. F/BTU)
1	352.81	0.537	1621.64	644.36	394020.61	1001.9	0.000140
7	347.48	0.525	1630.62	629.53	394020.61	1005.7	0.000143
11	336.74	0.514	1638.51	616.82	394020.61	1009.0	0.000129
15	352.81	0.543	1617.44	651.45	394020.61	1000.1	0.000368
21	325.92	0.499	1650.22	598.57	394020.61	1013.9	0.000658
26	331.35	0.524	1631.35	628.33	394020.61	1006.0	0.000308
31	325.92	0.485	1661.50	581.62	394020.61	1018.6	0.000145
37	298.47	0.452	1689.09	542.57	394020.61	1030.0	0.000229
41	315.01	0.472	1672.32	565.90	394020.61	1023.1	0.000277
47	352.81	0.543	1617.44	651.45	394020.61	1000.1	0.000358
57	347.48	0.531	1626.21	636.75	394020.61	1003.8	0.000484
71	342.12	0.527	1628.96	632.24	394020.61	1005.0	0.000305
75	320.48	0.504	1646.52	604.26	394020.61	1012.3	0.000609
103	331.35	0.506	1644.44	607.50	394020.61	1011.5	0.000405
108	304.01	0.456	1685.86	546.97	394020.61	1028.7	0.000456
113	342.12	0.533	1624.80	639.09	394020.61	1003.2	0.000286
118	342.12	0.520	1633.92	624.17	394020.61	1007.0	0.000358
128	331.35	0.512	1639.75	614.87	394020.61	1009.5	0.000410
138	325.92	0.502	1647.65	602.53	394020.61	1012.8	0.000481

Table B9: Sample Results Using Second Law of Thermodynamics for Exchanger Case 2

Day	Date	$\dot{S}_{gen,s}$ (W/ K)	$\dot{S}_{gen,t}$ (W/ K)	$\dot{S}_{gen,tot}$ (W/ K)	I_s (W)	I_t (W)	I_{total} (W)	Fouling resistance (hr. ft2. F/BTU)
1	10/15/99	32312.1	335290	367603	9.63E+06	9.99E+07	1.10E+08	0.00096
2	10/16/99	33912.2	367861	401773	1.01E+07	1.10E+08	1.20E+08	0.00109
3	10/17/99	33589.2	352818	386407	1.00E+07	1.05E+08	1.15E+08	0.00117
4	10/18/99	34551.1	365408	399959	1.03E+07	1.09E+08	1.19E+08	0.00098
5	10/19/99	34072.5	398755	432827	1.02E+07	1.19E+08	1.29E+08	0.00136
6	10/20/99	33967.0	393896	427863	1.01E+07	1.17E+08	1.28E+08	0.00125
7	10/21/99	32982.5	369361	402344	9.83E+06	1.10E+08	1.20E+08	0.00136
8	10/22/99	32707.6	396086	428794	9.75E+06	1.18E+08	1.28E+08	0.00150
9	10/23/99	32284.1	375113	407397	9.62E+06	1.12E+08	1.21E+08	0.00130
10	10/24/99	33503.9	414316	447820	9.98E+06	1.23E+08	1.33E+08	0.00142
11	10/25/99	32834.1	419060	451894	9.78E+06	1.25E+08	1.35E+08	0.00167
12	10/26/99	32130.5	394977	427108	9.57E+06	1.18E+08	1.27E+08	0.00173
13	10/27/99	34577.6	441309	475886	1.03E+07	1.32E+08	1.42E+08	0.00180
14	10/28/99	32789.0	453020	485809	9.77E+06	1.35E+08	1.45E+08	0.00215
15	10/29/99	31942.0	417934	449876	9.52E+06	1.25E+08	1.34E+08	0.00214
16	10/30/99	34186.7	487310	521497	1.02E+07	1.45E+08	1.55E+08	0.00231
17	10/31/99	32243.0	402261	434504	9.61E+06	1.20E+08	1.29E+08	0.00219
18	11/1/99	32627.0	437982	470609	9.72E+06	1.31E+08	1.40E+08	0.00258
19	11/2/99	34421.6	524222	558643	1.03E+07	1.56E+08	1.66E+08	0.00270
20	11/3/99	32688.0	483647	516335	9.74E+06	1.44E+08	1.54E+08	0.00303
21	11/4/99	34103.0	498115	532218	1.02E+07	1.48E+08	1.59E+08	0.00271
22	11/5/99	33433.9	497834	531268	9.96E+06	1.48E+08	1.58E+08	0.00294
23	11/6/99	34317.3	509553	543871	1.02E+07	1.52E+08	1.62E+08	0.00280
24	11/7/99	32723.6	507629	540352	9.75E+06	1.51E+08	1.61E+08	0.00315
25	11/8/99	33238.4	520451	553689	9.91E+06	1.55E+08	1.65E+08	0.00322
26	11/9/99	32584.4	511508	544092	9.71E+06	1.52E+08	1.62E+08	0.00336
27	11/10/99	34359.8	577589	611948	1.02E+07	1.72E+08	1.82E+08	0.00337
28	11/11/99	34129.0	611706	645835	1.02E+07	1.82E+08	1.92E+08	0.00357

Appendix C

Sample Results for From STX Program for Exchanger

Case 1

RESULTS OF DAY 31, TABLE A2 (SI UNITS)

HTC-STX	Version 3.5	Time: 3:49:12 PM		Date: 12/29/2001		File: BDFS_var	
*** Summary ***						SI Units	
Item No							
Service							
Calculation Mode	Rating Case						
Size	499 x 6090	Type	AEU - HORZ	Connections	3 Series		1 Parallel
Surface/Unit	279		Shells/unit	3		Surf/Shell	92.87
Cost/Unit	42,466		Cost/Surf	152.42		Weight/Shell	2,445
Heat Duty	796		MTD	15.45		F-corr	0.8730
Rate-Service	184.92		Calculated	341.93		Calc Fouling	0.00248

	Shell	Tubes				
Flow Rate	36151	30915	Tubes	19.050 x 2.108 on 25.40 30 deg		
Temperature In	-17.67	44.48	Tube No	250	Type:	PLAIN
Temperature Out	22.1	-3.9	Baffles:	VERT SEG	379.0 space	25.0 cut
Pressure Drop	16.324	14.029	Surface Area	OK. Over design by 84.91%		
Velocity	0.285	0.496	Shell pressure Drop	** Allowable exceeded.		
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.		
Film Coef.	967.8	695.2	Vibration	OK. Within limits.		
Nozzle In	1 x 101.6	101.6	Shell Nozzles	OK. Rho-V-Sqr within 6000		
Nozzle Out	1 x 101.6	101.6	Chan Nozzles	OK. Rho-V-Sqr within 9000		

RESULTS OF DAY 31, TABLE A2 (Cont.) (SI UNITS)

HTC-STX Version 3.5 Time: 3:49:12 PM Date: 12/29/2001 File: BDFS_var

*** Main *** SI Units

1	Job No		Item No.		RATING Case	
2	Case Desciption					
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series 1 Parallel
4	Size:	498.6 mm Dia	6,090.0 mm	Tube Length		mm Kettle Dia
5	Surface/Shell	m^2	93.3 Gross		92.9 Eff 2	U-Bend Area
6	Surface/Unit	m^2	279.8 Gross		278.6 Eff 6	U-Bend Area

	Performance of One Unit		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	kg/hr	36,150.9	30,915.3
9	Vapor	kg/hr	0.0	0.0
10	Liquid	kg/hr	36,150.9	30,915.3
11	Fluid Vap'z/Cond	kg/hr	0.0	0.0
12	Density In/Out	kg/m^3	752.000 / 752.000	800.000 / 800.000
13	Spec. Heat Vap/Liq	kJ/kg-C	0.000 / 2.052	0.000 / 1.855
14	Viscosity Vap/Liq	mPa.s	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	W/m-C	0.000 / 0.133	0.000 / 0.133
16	Temperature In/Out	°C	-17.7 / 22.14	44.5 / -3.93
17	Operating Pressure (Abs)	kPa	60.000	60.000
18	Press. Drop Allow/Calc	kPa	10.000 / 16.324	10.000 / 14.029
19	Number of Passes/Shell		1	2
20	Velocity, Average	m/sec	0.29	0.50
21	Film Coef.	W/m^2-C	967.80	695.20
22	Fouling Resist.	m^2-C/W	0.000000	0.000000

23	Heat Duty	796 kW		MTD/Wtd/Corr	15.45 °C	F-CORR	0.873	
24	Transfer Rate	184.92	Serv	341.93 Calc	341.93	Clean	0.00248	Foul

	Construction of One Shell					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	mm	494
27	Tube O.D	mm	19.050	No. Holes/TubeSheet		250
28	Tube I.D	mm	14.834	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	mm	25.4000
30	Tube Length Total	m	6.09	Tube Layout Angle		30
31	Tube Length Effective	m	6.06	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	mm	379
34	Window Area	m^2	0.0310	In/Out Spacing	mm	379.0/379.0
35	Seal Strips		YES	Drop Under Noz In/Out	mm	33.9/33.9

	Shell Nozzles		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	mm	101.60	101.60	Inside Dia.	mm	101.60	101.60
37	Velocity	m/sec	1.65	1.65	Velocity	m/sec	1.32	1.32
38	Rho-V-Sqr	kg/m-sec²	2036	2036	Rho-V-Sqr	kg/m-sec²	1400	1400
39	Nozzles/Shell	(OPP. SIDE)	1	1				

	Shellside Performance			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		6.737/4.163
41	Mass Vel Cross/Wind		214.8/323.9	Tubes		9.809
42	Mass Vel Long/Mean		81.0/263.8	Nozzles Shell/Tube		5.424/4.221
	Bundle Diameter Clearances			**Tube Metal Temperatures**		
43	Bundle-Shell	mm	5.08002	Avg. Tube Metal Temp.	°C	7.1
44	Baffle-Shell	mm	4.76250	Shellside Avg Surf. Temp	°C	6.9
45	Tube-Baffle	mm	0.92075	Tubeside Avg. Surf Temp	°C	7.2
46	Baffle Thk.	mm	6.350			

RESULTS OF DAY 31, TABLE A2 (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 3:47:39 PM		Date: 12/29/2001		File: BDFS_var	
*** Summary ***						English units	
Item No							
Service							
Calculation Mode	Rating Case						
Size	20 x 240	Type	AEU - HORZ	Connections	3 Series		1 Parallel
Surface/Unit	2,999		Shells/unit	3		Surf/Shell	999.69
Cost/Unit	42,466		Cost/Surf	14.16		Weight/Shell	5,390
Heat Duty	2,715,491		MTD	27.80		F-corr	0.8730
Rate-Service	32.57		Calculated	60.22		Calc Fouling	0.01410

	Shell	Tubes			
	Shell	**Tubes**	Tubes	0.750 x 0.083 on 1.00 30 deg	
Flow Rate	79698	68155	Tube No	250 Type:	PLAIN
Temperature In	0.19	112.06	Baffles:	VERT SEG 14.9 space	25.0 cut
Temperature Out	71.9	24.9			
Pressure Drop	2.370	2.037	Surface Area	OK. Over design by 84.9%	
Velocity	0.937	1.630	Shell pressure Drop	** Allowable exceeded.	
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.	
Film Coef.	170.4	122.4	Vibration	OK. Within limits.	
Nozzle In	1 x 4.0	4.0	Shell Nozzles	OK. Rho-V-Sqr within 4000	
Nozzle Out	1 x 4.0	4.0	Chan Nozzles	OK. Rho-V-Sqr within 6000	

RESULTS OF DAY 31, TABLE A2 (Cont.) (ENGLISH UNITS)

	HTC-STX	Version 3.5	Time: 3:47:39 PM		Date: 12/29/2001		File: BDFS_var
	*** Main ***						English units
1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	19.630 in Dia	239.8 in	Tube Length		in	Kettle Dia
5	Surface/Shell	ft²	1,003.9	Gross	999.7 Eff 23		U-Bend Area
6	Surface/Unit	ft²	3,011.7	Gross	2,999.1 Eff 69		U-Bend Area
	Performance of One Unit			**SHELLSIDE**			**TUBESIDE**
7	Fluid Circulated			shell side			tube side
8	Total Fluid In	lb/hr		79,697.7			68,155.4
9	Vapor	lb/hr		0.0			0.0
10	Liquid	lb/hr		79,697.7			68,155.4
11	Fluid Vap'z/Cond	lb/hr		0.0			0.0
12	Density In/Out	lb/ft³		46.946 / 46.946			49.942 / 49.942
13	Spec. Heat Vap/Liq	Btu/lb-F		0.000 / 0.490			0.000 / 0.443
14	Viscosity Vap/Liq	cP		0.000 / 0.546			0.000 / 0.678
15	Therm Cond Vap/Liq	Btu/hr-ft-F		0.000 / 0.077			0.000 / 0.077
16	Temperature In/Out	°F		0.2 / 71.85			112.1 / 24.93
17	Operating Pressure (Abs)	psi		8.702			8.702
18	Press. Drop Allow/Calc	psi		1.450 / 2.370			1.450 / 2.037
19	Number of Passes/Shell			1			2
20	Velocity, Average	ft/sec		0.94			1.63
21	Film Coef.	Btu/hr-ft²-F		170.45			122.44
22	Fouling Resist.	hr-ft²-F/Btu		0.000000			0.000000
23	Heat Duty	2,715,491 Btu/hr	MTD/Wtd/Corr		27.80 °F	F-CORR	0.873
24	Transfer Rate	32.57 Serv	60.22	Calc	60.22	Clean	0.01410 Foul
	Construction of One Shell						
25	TEMA Shell Type		E	Rear End Type			U.T.
26	Tube Type		PLAIN	Bundle Dia	in		19.43
27	Tube O.D	in	0.750	No. Holes/TubeSheet			250
28	Tube I.D	in	0.584	No. Holes Counted			262
29	Area Ratio		1.284	Tube Pitch	in		1.0000
30	Tube Length Total	ft	19.98	Tube Layout Angle			30
31	Tube Length Effective	ft	19.89	Impingement Plate			YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell			16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	in		14.921
34	Window Area	in²	48.0492	In/Out Spacing	in		14.9/14.9
35	Seal Strips		YES	Drop Under Noz In/Out	in		1.3/1.3
	Shell Nozzles	**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia, in	4.00	4.00	Inside Dia,	in	4.00	4.00
37	Velocity ft/sec	5.40	5.40	Velocity	ft/sec	4.34	4.34
38	Rho-V-Sqr lb/ft-sec²	1371	1371	Rho-V-Sqr	lb/ft-sec²	942	942
39	Nozzles/Shell (OPP. SIDE)	1	1				
	Shellside Performance			**Pressure Drop**			
40	Bundle Flow Fraction		0.786	Shell Cross/Wind			0.978/0.604
41	Mass Vel Cross/Wind		44.0/66.3	Tubes			1.424
42	Mass Vel Long/Mean		16.6/54.0	Nozzles Shell/Tube			0.788/0.613
	Bundle Diameter Clearances			**Tube Metal Temperatures**			
43	Bundle-Shell	in	0.20000	Avg. Tube Metal Temp.	°F		44.7
44	Baffle-Shell	in	0.18750	Shellside Avg Surf. Temp	°F		44.5
45	Tube-Baffle	in	0.03625	Tubeside Avg. Surf Temp	°F		45.0
46	Baffle Thk.	in	0.250				

RESULTS OF DAY 51, TABLE A2 (SI UNITS)

HTC-STX	Version 3.5	Time: 3:58:09 PM	Date: 12/29/2001	File: BDFS_var	
*** Summary ***				SI Units	
Item No					
Service					
Calculation Mode	Rating Case				
Size	499 x 6090	Type	AEU - HORZ Connections	3 Series	1 Parallel
Surface/Unit	279	Shells/unit	3	Surf/Shell	92.87
Cost/Unit	42,466	Cost/Surf	152.42	Weight/Shell	2,445
Heat Duty	835	MTD	18.25	F-corr	0.9014
Rate-Service	164.14	Calculated	352.73	Calc Fouling	0.00326

	Shell	Tubes		
			Tubes	19.050 x 2.108 on 25.40 30 deg
Flow Rate	36532	32292	Tube No	250 Type: PLAIN
Temperature In	-17.84	47.59	Baffles:	VERT SEG 379.0 space 25.0 cut
Temperature Out	23.5	-1.0		
Pressure Drop	16.659	15.203	Surface Area	OK. Over design by 114.9%
Velocity	0.288	0.518	Shell pressure Drop	** Allowable exceeded.
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.
Film Coef.	973.9	727.9	Vibration	OK. Within limits.
Nozzle In	1 x 101.6	101.6	Shell Nozzles	OK. Rho-V-Sqr within 6000
Nozzle Out	1 x 101.6	101.6	Chan Nozzles	OK. Rho-V-Sqr within 9000

RESULTS OF DAY 51, TABLE A2 (Cont.) (SI UNITS)

HTC-STX Version 3.5 Time: 3:58:09 PM Date: 12/29/2001 File: BDFS_var

***** Main ***** SI Units

1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	498.6 mm	Dia	6,090.0 mm	Tube Length	mm	Kettle Dia
5	Surface/Shell	m²	93.3	Gross	92.9 Eff 2		U-Bend Area
6	Surface/Unit	m²	279.8	Gross	278.6 Eff 6		U-Bend Area

	Performance of One Unit		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	kg/hr	36,532.3	32,292.4
9	Vapor	kg/hr	0.0	0.0
10	Liquid	kg/hr	36,532.3	32,292.4
11	Fluid Vap'z/Cond	kg/hr	0.0	0.0
12	Density In/Out	kg/m³	752.000 / 752.000	800.000 / 800.000
13	Spec. Heat Vap/Liq	kJ/kg-C	0.000 / 2.052	0.000 / 1.855
14	Viscosity Vap/Liq	mPa.s	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	W/m-C	0.000 / 0.133	0.000 / 0.133
16	Temperature In/Out	°C	-17.8 / 23.48	47.6 / -1.02
17	Operating Pressure (Abs)	kPa	60.000	60.000
18	Press. Drop Allow/Calc	kPa	10.000 / 16.659	10.000 / 15.203
19	Number of Passes/Shell		1	2
20	Velocity, Average	m/sec	0.29	0.52
21	Film Coef.	W/m²-C	973.91	727.93
22	Fouling Resist.	m²-C/W	0.000000	0.000000

23	Heat Duty	835 kW		MTD/Wtd/Corr		18.25 °C	F-CORR	0.901	
24	Transfer Rate	164.14	Serv	352.73	Calc	352.73	Clean	0.00326	Foul

	Construction of One Shell					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	mm	494
27	Tube O.D	mm	19.050	No. Holes/TubeSheet		250
28	Tube I.D	mm	14.834	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	mm	25.4000
30	Tube Length Total	m	6.09	Tube Layout Angle		30
31	Tube Length Effective	m	6.06	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	mm	379
34	Window Area	m²	0.0310	In/Out Spacing	mm	379.0/379.0
35	Seal Strips		YES	Drop Under Noz In/Out	mm	33.9/33.9

	Shell Nozzles		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	mm	101.60	101.60	Inside Dia.	mm	101.60	101.60
37	Velocity	m/sec	1.66	1.66	Velocity	m/sec	1.38	1.38
38	Rho-V-Sqr	kg/m-sec²	2079	2079	Rho-V-Sqr	kg/m-sec²	1527	1527
39	Nozzles/Shell	(OPP. SIDE)	1	1				

	Shellside Performance			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		6.868/4.251
41	Mass Vel Cross/Wind		217.1/327.3	Tubes		10.598
42	Mass Vel Long/Mean		81.8/266.6	Nozzles Shell/Tube		5.539/4.605
	Bundle Diameter Clearances			**Tube Metal Temperatures**		
43	Bundle-Shell	mm	5.08002	Avg. Tube Metal Temp.	°C	9.1
44	Baffle-Shell	mm	4.76250	Shellside Avg Surf. Temp	°C	9.0
45	Tube-Baffle	mm	0.92075	Tubeside Avg. Surf Temp	°C	9.3
46	Baffle Thk.	mm	6.350			

RESULTS OF DAY 51, TABLE A2 (ENGLISH UNITS)

HTC-STX *** Summary ***	Version 3.5		Time: 3:59:21 PM	Date: 12/29/2001		File: BDFS_var English units
Item No						
Service						
Calculation Mode	Rating Case					
Size	20 x 240	Type	AEU - HORZ	Connections	3 Series	1 Parallel
Surface/Unit	2,999		Shells/unit	3	Surf/Shell	999.69
Cost/Unit	42,466		Cost/Surf	14.16	Weight/Shell	5,390
Heat Duty	2,848,201		MTD	32.85	F-corr	0.9014
Rate-Service	28.91		Calculated	62.12	Calc Fouling	0.01850

	Shell	Tubes				
			Tubes	0.750 x 0.083 on 1.00 30 deg		
Flow Rate	80539	71191	Tube No	250	Type:	PLAIN
Temperature In	-0.11	117.66	Baffles:	VERT SEG	14.9 space	25.0 cut
Temperature Out	74.3	30.2				
Pressure Drop	2.419	2.207	Surface Area		OK. Over design by 114.89%	
Velocity	0.947	1.703	Shell pressure Drop		** Allowable exceeded.	
Passes	1	2	Tube Pressure Drop		** Allowable exceeded.	
Film Coef.	171.5	128.2	Vibration		OK. Within limits.	
Nozzle In	1 x 4.0	4.0	Shell Nozzles		OK. Rho-V-Sqr within 4000	
Nozzle Out	1 x 4.0	4.0	Chan Nozzles		OK. Rho-V-Sqr within 6000	

RESULTS OF DAY 51, TABLE A2 (Cont.) (ENGLISH UNITS)

HTC-STX Version 3.5 Time: 3:59:21 PM Date: 12/29/2001 File: BDFS_var
***** Main ***** English units

1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	19.630 in Dia	239.8 in	Tube Length		in	Kettle Dia
5	Surface/Shell	ft²	1,003.9	Gross	999.7 Eff 23		U-Bend Area
6	Surface/Unit	ft²	3,011.7	Gross	2,999.1 Eff 69		U-Bend Area

	Performance of One Unit		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	lb/hr	80,538.6	71,191.4
9	Vapor	lb/hr	0.0	0.0
10	Liquid	lb/hr	80,538.6	71,191.4
11	Fluid Vap'z/Cond	lb/hr	0.0	0.0
12	Density In/Out	lb/ft³	46.946 / 46.946	49.942 / 49.942
13	Spec. Heat Vap/Liq	Btu/lb-F	0.000 / 0.490	0.000 / 0.443
14	Viscosity Vap/Liq	cP	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	Btu/hr-ft-F	0.000 / 0.077	0.000 / 0.077
16	Temperature In/Out	°F	-0.1 / 74.26	117.7 / 30.16
17	Operating Pressure (Abs)	psi	8.702	8.702
18	Press. Drop Allow/Calc	psi	1.450 / 2.419	1.450 / 2.207
19	Number of Passes/Shell		1	2
20	Velocity, Average	ft/sec	0.95	1.70
21	Film Coef.	Btu/hr-ft²-F	171.53	128.20
22	Fouling Resist.	hr-ft²-F/Btu	0.000000	0.000000

23	Heat Duty	2,848,201 Btu/hr	MTD/Wtd/Corr		32.85 °F	F-CORR	0.901	
24	Transfer Rate	28.91	Serv 62.12	Calc	62.12	Clean	0.01850	Foul

	Construction of One Shell					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	in	19.43
27	Tube O.D	in	0.750	No. Holes/TubeSheet		250
28	Tube I.D	in	0.584	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	in	1.0000
30	Tube Length Total	ft	19.98	Tube Layout Angle		30
31	Tube Length Effective	ft	19.89	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	in	14.921
34	Window Area	in²	48.0492	In/Out Spacing	in	14.9/14.9
35	Seal Strips		YES	Drop Under Noz In/Out	in	1.3/1.3

	Shell Nozzles		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	in	4.00	4.00	Inside Dia.	in	4.00	4.00
37	Velocity	ft/sec	5.46	5.46	Velocity	ft/sec	4.54	4.54
38	Rho-V-Sqr	lb/ft-sec²	1400	1400	Rho-V-Sqr	lb/ft-sec²	1028	1028
39	Nozzles/Shell	(OPP. SIDE)	1	1				

	Shellside Performance		**Pressure Drop**	
40	Bundle Flow Fraction	0.786	Shell Cross/Wind	0.997/0.617
41	Mass Vel Cross/Wind	44.5/67.0	Tubes	1.539
42	Mass Vel Long/Mean	16.8/54.6	Nozzles Shell/Tube	0.804/0.669

	Bundle Diameter Clearances			**Tube Metal Temperatures**		
43	Bundle-Shell	in	0.20000	Avg. Tube Metal Temp.	°F	48.4
44	Baffle-Shell	in	0.18750	Shellside Avg Surf. Temp	°F	48.1
45	Tube-Baffle	in	0.03625	Tubeside Avg. Surf Temp	°F	48.7
46	Baffle Thk.	in	0.250			

RESULTS OF DAY 65, TABLE A2 (SI UNITS)

HTC-STX	Version 3.5	Time: 2:20:49 PM		Date: 12/30/2001		File: BDFS_var	
*** Summary ***						SI Units	
Item No							
Service							
Calculation Mode	Rating Case						
Size	499 x 6090	Type	AEU - HORZ	Connections	3 Series		1 Parallel
Surface/Unit	279		Shells/unit	3		Surf/Shell	92.87
Cost/Unit	42,466		Cost/Surf	152.42		Weight/Shell	2,445
Heat Duty	798		MTD	17.87		F-corr	0.9012
Rate-Service	160.28		Calculated	347.01		Calc Fouling	0.00336

	Shell	Tubes				
			Tubes	19.050 x 2.108 on 25.40 30 deg		
Flow Rate	35104	31912	Tube No	250	Type:	PLAIN
Temperature In	-17.8	46.23	Baffles:	VERT SEG	379.0 space	25.0 cut
Temperature Out	23.3	-0.8				
Pressure Drop	15.422	14.874	Surface Area		OK. Over design by 116.5%	
Velocity	0.277	0.512	Shell pressure Drop		** Allowable exceeded.	
Passes	1	2	Tube Pressure Drop		** Allowable exceeded.	
Film Coef.	950.9	719.0	Vibration		OK. Within limits.	
Nozzle In	1 x 101.6	101.6	Shell Nozzles		OK. Rho-V-Sqr within 6000	
Nozzle Out	1 x 101.6	101.6	Chan Nozzles		OK. Rho-V-Sqr within 9000	

RESULTS OF DAY 65, TABLE A2 (Cont.) (SI UNITS)

HTC-STX	Version 3.5	Time: 2:20:49 PM	Date: 12/30/2001	File: BDFS_var
*** Main ***				SI Units

No.								
1	Job No			Item No.		RATING Case		
2	Case Desciption							
3	TEMA Type	AEU - HORZ		Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	498.6 mm	Dia	6,090.0 mm	Tube Length		mm	Kettle Dia
5	Surface/Shell	m^2	93.3	Gross		92.9 Eff 2		U-Bend Area
6	Surface/Unit	m^2	279.8	Gross		278.6 Eff 6		U-Bend Area

No.	***Performance of One Unit***		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	kg/hr	35,103.8	31,912.0
9	Vapor	kg/hr	0.0	0.0
10	Liquid	kg/hr	35,103.8	31,912.0
11	Fluid Vap'z/Cond	kg/hr	0.0	0.0
12	Density In/Out	kg/m^3	752.000 / 752.000	800.001 / 800.001
13	Spec. Heat Vap/Liq	kJ/kg-C	0.000 / 2.052	0.000 / 1.855
14	Viscosity Vap/Liq	mPa.s	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	W/m-C	0.000 / 0.133	0.000 / 0.133
16	Temperature In/Out	°C	-17.8 / 23.27	46.2 / -0.8
17	Operating Pressure (Abs)	kPa	60.000	60.000
18	Press. Drop Allow/Calc	kPa	10.000 / 15.422	10.000 / 14.874
19	Number of Passes/Shell		1	2
20	Velocity, Average	m/sec	0.28	0.51
21	Film Coef.	W/m^2-C	950.88	719.01
22	Fouling Resist.	m^2-C/W	0.000000	0.000000

No.									
23	Heat Duty	798 kW		MTD/Wtd/Corr		17.87 °C	F-CORR	0.901	
24	Transfer Rate	160.28	Serv	347.01	Calc	347.01	Clean	0.00336	Foul

No.	***Construction of One Shell***					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	mm	494
27	Tube O.D	mm	19.050	No. Holes/TubeSheet		250
28	Tube I.D	mm	14.834	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	mm	25.4000
30	Tube Length Total	m	6.09	Tube Layout Angle		30
31	Tube Length Effective	m	6.06	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	mm	379
34	Window Area	m^2	0.0310	In/Out Spacing	mm	379.0/379.0
35	Seal Strips		YES	Drop Under Noz In/Out	mm	33.9/33.9

No.	**Shell Nozzles**		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	mm	101.60	101.60	Inside Dia.	mm	101.60	101.60
37	Velocity	m/sec	1.60	1.60	Velocity	m/sec	1.37	1.37
38	Rho-V-Sqr	kg/m-sec²	1920	1920	Rho-V-Sqr	kg/m-sec²	1491	1491
39	Nozzles/Shell	(OPP. SIDE)	1	1				

No.	**Shellside Performance**			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		6.382/3.925
41	Mass Vel Cross/Wind		208.6/314.5	Tubes		10.377
42	Mass Vel Long/Mean		78.6/256.1	Nozzles Shell/Tube		5.114/4.497

No.	**Bundle Diameter Clearances**			**Tube Metal Temperatures**		
43	Bundle-Shell	mm	5.08002	Avg. Tube Metal Temp.	°C	9.1
44	Baffle-Shell	mm	4.76250	Shellside Avg Surf. Temp	°C	9.0
45	Tube-Baffle	mm	0.92075	Tubeside Avg. Surf Temp	°C	9.3
46	Baffle Thk.	mm	6.350			

RESULTS OF DAY 65, TABLE A2 (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 2:21:02 PM	Date: 12/30/2001	File: BDFS_var	
*** Summary ***				English units	
Item No					
Service					
Calculation Mode	Rating Case				
Size	20 x 240	Type	AEU - HORZ Connections	3 Series	1 Parallel
Surface/Unit	2,999	Shells/unit	3	Surf/Shell	999.69
Cost/Unit	42,466	Cost/Surf	14.16	Weight/Shell	5,390
Heat Duty	2,723,377	MTD	32.17	F-corr	0.9012
Rate-Service	28.23	Calculated	61.11	Calc Fouling	0.01906

	Shell	Tubes		
	Shell	**Tubes**	Tubes	0.750 x 0.083 on 1.00 30 deg
Flow Rate	77389	70353	Tube No	250 Type: PLAIN
Temperature In	-0.04	115.21	Baffles:	VERT SEG 14.9 space 25.0 cut
Temperature Out	73.9	30.6		
Pressure Drop	2.239	2.160	Surface Area	OK. Over design by 116.5%
Velocity	0.910	1.683	Shell pressure Drop	** Allowable exceeded.
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.
Film Coef.	167.5	126.6	Vibration	OK. Within limits.
Nozzle In	1 x 4.0	4.0	Shell Nozzles	OK. Rho-V-Sqr within 4000
Nozzle Out	1 x 4.0	4.0	Chan Nozzles	OK. Rho-V-Sqr within 6000

RESULTS OF DAY 65, TABLE A2 (Cont.) (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 2:21:02 PM	Date: 12/30/2001	File: BDFS_var	
*** Main ***				English units	

#							
1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	19.630 in Dia	239.8 in	Tube Length		in	Kettle Dia
5	Surface/Shell	ft^2	1,003.9	Gross	999.7 Eff 23		U-Bend Area
6	Surface/Unit	ft^2	3,011.7	Gross	2,999.1 Eff 69		U-Bend Area

#	***Performance of One Unit***		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	lb/hr	77,389.3	70,352.7
9	Vapor	lb/hr	0.0	0.0
10	Liquid	lb/hr	77,389.3	70,352.7
11	Fluid Vap'z/Cond	lb/hr	0.0	0.0
12	Density In/Out	lb/ft^3	46.946 / 46.946	49.942 / 49.942
13	Spec. Heat Vap/Liq	Btu/lb-F	0.000 / 0.490	0.000 / 0.443
14	Viscosity Vap/Liq	cP	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	Btu/hr-ft-F	0.000 / 0.077	0.000 / 0.077
16	Temperature In/Out	°F	0.0 / 73.89	115.2 / 30.56
17	Operating Pressure (Abs)	psi	8.702	8.702
18	Press. Drop Allow/Calc	psi	1.450 / 2.239	1.450 / 2.160
19	Number of Passes/Shell		1	2
20	Velocity, Average	ft/sec	0.91	1.68
21	Film Coef.	Btu/hr-ft^2-F	167.47	126.63
22	Fouling Resist.	hr-ft^2-F/Btu	0.000000	0.000000

#									
23	Heat Duty	2,723,377	Btu/hr	MTD/Wtd/Corr		32.17	°F	F-CORR	0.901
24	Transfer Rate	28.23	Serv	61.11	Calc	61.11	Clean	0.01906	Foul

#	***Construction of One Shell***					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	in	19.43
27	Tube O.D	in	0.750	No. Holes/TubeSheet		250
28	Tube I.D	in	0.584	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	in	1.0000
30	Tube Length Total	ft	19.98	Tube Layout Angle		30
31	Tube Length Effective	ft	19.89	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	in	14.921
34	Window Area	in^2	48.0492	In/Out Spacing	in	14.9/14.9
35	Seal Strips		YES	Drop Under Noz In/Out	in	1.3/1.3

#	**Shell Nozzles**		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	in	4.00	4.00	Inside Dia.	in	4.00	4.00
37	Velocity	ft/sec	5.25	5.25	Velocity	ft/sec	4.48	4.48
38	Rho-V-Sqr	lb/ft-sec^2	1293	1293	Rho-V-Sqr	lb/ft-sec^2	1004	1004
39	Nozzles/Shell	(OPP. SIDE)	1	1				

#	**Shellside Performance**			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		0.927/0.570
41	Mass Vel Cross/Wind		42.7/64.4	Tubes		1.507
42	Mass Vel Long/Mean		16.1/52.5	Nozzles Shell/Tube		0.743/0.653

#	**Bundle Diameter Clearances**			**Tube Metal Temperatures**		
43	Bundle-Shell	in	0.20000	Avg. Tube Metal Temp.	°F	48.4
44	Baffle-Shell	in	0.18750	Shellside Avg Surf. Temp	°F	48.2
45	Tube-Baffle	in	0.03625	Tubeside Avg. Surf Temp	°F	48.7
46	Baffle Thk.	in	0.250			

RESULTS OF DAY 83, TABLE A2 (SI UNITS)

HTC-STX	Version 3.5	Time: 4:04:43 PM		Date: 12/29/2001		File: BDFS_var	
*** Summary ***						SI Units	
Item No							
Service							
Calculation Mode	Rating Case						
Size	499 x 6090	Type	AEU - HORZ	Connections	3 Series	1 Parallel	
Surface/Unit	279		Shells/unit	3		Surf/Shell	92.87
Cost/Unit	42,466		Cost/Surf	152.42		Weight/Shell	2,445
Heat Duty	799		MTD	18.03		F-corr	0.9026
Rate-Service	159.03		Calculated	347.52		Calc Fouling	0.00341

	Shell	Tubes				
	Shell	**Tubes**	Tubes	19.050 x 2.108 on 25.40 30 deg		
Flow Rate	35040	32006	Tube No	250	Type:	PLAIN
Temperature In	-18.07	46.14	Baffles:	VERT SEG	379.0 space	25.0 cut
Temperature Out	23.2	-0.8				
Pressure Drop	15.368	14.955	Surface Area		OK. Over design by 118.51%	
Velocity	0.277	0.514	Shell pressure Drop		** Allowable exceeded.	
Passes	1	2	Tube Pressure Drop		** Allowable exceeded.	
Film Coef.	949.8	721.2	Vibration		OK. Within limits.	
Nozzle In	1 x 101.6	101.6	Shell Nozzles		OK. Rho-V-Sqr within 6000	
Nozzle Out	1 x 101.6	101.6	Chan Nozzles		OK. Rho-V-Sqr within 9000	

RESULTS OF DAY 83, TABLE A2 (Cont.) (SI UNITS)

HTC-STX	Version 3.5	Time: 4:04:43 PM	Date: 12/29/2001	File: BDFS_var
*** Main ***				SI Units

1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	498.6 mm Dia	6,090.0 mm	Tube Length		mm	Kettle Dia
5	Surface/Shell	m^2	93.3	Gross	92.9 Eff 2		U-Bend Area
6	Surface/Unit	m^2	279.8	Gross .	278.6 Eff 6		U-Bend Area

	Performance of One Unit		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	kg/hr	35,039.7	32,006.3
9	Vapor	kg/hr	0.0	0.0
10	Liquid	kg/hr	35,039.7	32,006.3
11	Fluid Vap'z/Cond	kg/hr	0.0	0.0
12	Density In/Out	kg/m^3	752.000 / 752.000	800.000 / 800.000
13	Spec. Heat Vap/Liq	kJ/kg-C	0.000 / 2.052	0.000 / 1.855
14	Viscosity Vap/Liq	mPa.s	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	W/m-C	0.000 / 0.133	0.000 / 0.133
16	Temperature In/Out	°C	-18.1 / 23.17	46.1 / -0.81
17	Operating Pressure (Abs)	kPa	60.000	60.000
18	Press. Drop Allow/Calc	kPa	10.000 / 15.368	10.000 / 14.955
19	Number of Passes/Shell		1	2
20	Velocity, Average	m/sec	0.28	0.51
21	Film Coef.	W/m^2-C	949.84	721.20
22	Fouling Resist.	m^2-C/W	0.000000	0.000000

23	Heat Duty	799 kW		MTD/Wtd/Corr		18.03 °C	F-CORR	0.903	
24	Transfer Rate	159.03	Serv	347.52	Calc	347.52	Clean	0.00341	Foul

	Construction of One Shell							
25	TEMA Shell Type			E	Rear End Type			U.T.
26	Tube Type			PLAIN	Bundle Dia	mm		494
27	Tube O.D	mm		19.050	No. Holes/TubeSheet			250
28	Tube I.D	mm		14.834	No. Holes Counted			262
29	Area Ratio			1.284	Tube Pitch	mm		25.4000
30	Tube Length Total	m		6.09	Tube Layout Angle			30
31	Tube Length Effective	m		6.06	Impingement Plate			YES
32	Baffle Type			VERT- SEG	Crosspasses/Shell			16
33	Baffle Cut, Frac Dia/NFA			0.340/0.250	Central Spacing	mm		379
34	Window Area	m^2		0.0310	In/Out Spacing	mm		379.0/379.0
35	Seal Strips			YES	Drop Under Noz In/Out	mm		33.9/33.9
	Shell Nozzles		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia,	mm	101.60	101.60	Inside Dia,	mm	101.60	101.60
37	Velocity	m/sec	1.59	1.59	Velocity	m/sec	1.37	1.37
38	Rho-V-Sqr	$kg/m\text{-}sec^2$	1913	1913	Rho-V-Sqr	$kg/m\text{-}sec^2$	1500	1500
39	Nozzles/Shell	(OPP. SIDE)	1	1				
	Shellside Performance				**Pressure Drop**			
40	Bundle Flow Fraction			0.786	Shell Cross/Wind			6.361/3.911
41	Mass Vel Cross/Wind			208.2/314.0	Tubes			10.432
42	Mass Vel Long/Mean			78.5/255.7	Nozzles Shell/Tube			5.096/4.524
	Bundle Diameter Clearances				**Tube Metal Temperatures**			
43	Bundle-Shell	mm		5.08002	Avg. Tube Metal Temp.	°C		9.1
44	Baffle-Shell	mm		4.76250	Shellside Avg Surf. Temp	°C		8.9
45	Tube-Baffle	mm		0.92075	Tubeside Avg. Surf Temp	°C		9.2
46	Baffle Thk.	mm		6.350				

RESULTS OF DAY 83, TABLE A2 (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 4:05:11 PM		Date: 12/29/2001		File: BDFS_var	
*** Summary ***						English units	
Item No							
Service							
Calculation Mode	Rating Case						
Size	20 x 240	Type	AEU - HORZ	Connections	3 Series		1 Parallel
Surface/Unit	2,999		Shells/unit	3		Surf/Shell	999.69
Cost/Unit	42,466		Cost/Surf	14.16		Weight/Shell	5,390
Heat Duty	2,726,554		MTD	32.46		F-corr	0.9026
Rate-Service	28.01		Calculated	61.21		Calc Fouling	0.01936

	Shell	Tubes		
	Shell	**Tubes**	Tubes	0.750 x 0.083 on 1.00 30 deg
Flow Rate	77248	70561	Tube No	250 Type: PLAIN
Temperature In	-0.53	115.05	Baffles:	VERT SEG 14.9 space 25.0 cut
Temperature Out	73.7	30.5		
Pressure Drop	2.231	2.171	Surface Area	OK. Over design by 118.51%
Velocity	0.908	1.688	Shell pressure Drop	** Allowable exceeded.
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.
Film Coef.	167.3	127.0	Vibration	OK. Within limits.
Nozzle In	1 x 4.0	4.0	Shell Nozzles	OK. Rho-V-Sqr within 4000
Nozzle Out	1 x 4.0	4.0	Chan Nozzles	OK. Rho-V-Sqr within 6000

RESULTS OF DAY 83, TABLE A2 (Cont.) (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 4:05:11 PM	Date: 12/29/2001	File: BDFS_var
*** Main ***				English units

#							
1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	19.630 in Dia		239.8 in Tube Length		in	Kettle Dia
5	Surface/Shell	ft²	1,003.9	Gross	999.7 Eff 23		U-Bend Area
6	Surface/Unit	ft²	3,011.7	Gross	2,999.1 Eff 69		U-Bend Area

#	*Performance of One Unit*		SHELLSIDE	TUBESIDE
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	lb/hr	77,248.1	70,560.6
9	Vapor	lb/hr	0.0	0.0
10	Liquid	lb/hr	77,248.1	70,560.6
11	Fluid Vap'z/Cond	lb/hr	0.0	0.0
12	Density In/Out	lb/ft³	46.946 / 46.946	49.942 / 49.942
13	Spec. Heat Vap/Liq	Btu/lb-F	0.000 / 0.490	0.000 / 0.443
14	Viscosity Vap/Liq	cP	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	Btu/hr-ft-F	0.000 / 0.077	0.000 / 0.077
16	Temperature In/Out	°F	-0.5 / 73.71	115.1 / 30.54
17	Operating Pressure (Abs)	psi	8.702	8.702
18	Press. Drop Allow/Calc	psi	1.450 / 2.231	1.450 / 2.171
19	Number of Passes/Shell		1	2
20	Velocity, Average	ft/sec	0.91	1.69
21	Film Coef.	Btu/hr-ft²-F	167.29	127.02
22	Fouling Resist.	hr-ft²-F/Btu	0.000000	0.000000

#									
23	Heat Duty	2,726,554	Btu/hr	MTD/Wtd/Corr		32.46 °F	F-CORR	0.903	
24	Transfer Rate	28.01	Serv	61.21	Calc	61.21	Clean	0.01936	Foul

Construction of One Shell

#						
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	in	19.43
27	Tube O.D	in	0.750	No. Holes/TubeSheet		250
28	Tube I.D	in	0.584	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	in	1.0000
30	Tube Length Total	ft	19.98	Tube Layout Angle		30
31	Tube Length Effective	ft	19.89	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	in	14.921
34	Window Area	in²	48.0492	In/Out Spacing	in	14.9/14.9
35	Seal Strips		YES	Drop Under Noz In/Out	in	1.3/1.3

#	**Shell Nozzles**		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	in	4.00	4.00	Inside Dia.	in	4.00	4.00
37	Velocity	ft/sec	5.24	5.24	Velocity	ft/sec	4.50	4.50
38	Rho-V-Sqr	lb/ft-sec²	1288	1288	Rho-V-Sqr	lb/ft-sec²	1010	1010
39	Nozzles/Shell	(OPP. SIDE)	1	1				

#	**Shellside Performance**			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		0.924/0.568
41	Mass Vel Cross/Wind		42.6/64.3	Tubes		1.515
42	Mass Vel Long/Mean		16.1/52.4	Nozzles Shell/Tube		0.740/0.657
	Bundle Diameter Clearances			**Tube Metal Temperatures**		
43	Bundle-Shell	in	0.20000	Avg. Tube Metal Temp.	°F	48.3
44	Baffle-Shell	in	0.18750	Shellside Avg Surf. Temp	°F	48.1
45	Tube-Baffle	in	0.03625	Tubeside Avg. Surf Temp	°F	48.6
46	Baffle Thk.	in	0.250			

RESULTS OF DAY 97, TABLE A2 (SI UNITS)

HTC-STX *** Summary ***	Version 3.5		Time: 4:10:17 PM	Date: 12/29/2001		File: BDFS_var SI Units
Item No						
Service						
Calculation Mode	Rating Case					
Size	499 x 6090	Type	AEU - HORZ	Connections	3 Series	1 Parallel
Surface/Unit	279		Shells/unit	3	Surf/Shell	92.87
Cost/Unit	42,466		Cost/Surf	152.42	Weight/Shell	2,445
Heat Duty	739		MTD	17.66	F-corr	0.8979
Rate-Service	150.15		Calculated	323.45	Calc Fouling	0.00357

	Shell	Tubes		
	Shell	**Tubes**	Tubes	19.050 x 2.108 on 25.40 30 deg
Flow Rate	32355	29417	Tube No	250 Type: PLAIN
Temperature In	-17.8	46.35	Baffles:	VERT SEG 379.0 space 25.0 cut
Temperature Out	23.7	-0.9		
Pressure Drop	13.174	12.803	Surface Area	OK. Over design by 115.41%
Velocity	0.255	0.472	Shell pressure Drop	** Allowable exceeded.
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.
Film Coef.	905.5	660.9	Vibration	OK. Within limits.
Nozzle In	1 x 101.6	101.6	Shell Nozzles	OK. Rho-V-Sqr within 6000
Nozzle Out	1 x 101.6	101.6	Chan Nozzles	OK. Rho-V-Sqr within 9000

RESULTS OF DAY 97, TABLE A2 (Cont.) (SI UNITS)

	HTC-STX	Version 3.5	Time: 4:10:17 PM		Date: 12/29/2001		File: BDFS_var
	***** Main *****						SI Units
1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	498.6 mm Dia	6,090.0 mm	Tube Length		mm	Kettle Dia
5	Surface/Shell	m²	93.3	Gross	92.9 Eff 2		U-Bend Area
6	Surface/Unit	m²	279.8	Gross	278.6 Eff 6		U-Bend Area

	Performance of One Unit		**SHELLSIDE**	**TUBESIDE**
7	Fluid Circulated		shell side	tube side
8	Total Fluid In	kg/hr	32,354.9	29,416.7
9	Vapor	kg/hr	0.0	0.0
10	Liquid	kg/hr	32,354.9	29,416.7
11	Fluid Vap'z/Cond	kg/hr	0.0	0.0
12	Density In/Out	kg/m³	752.000 / 752.000	800.000 / 800.000
13	Spec. Heat Vap/Liq	kJ/kg-C	0.000 / 2.052	0.000 / 1.855
14	Viscosity Vap/Liq	mPa.s	0.000 / 0.546	0.000 / 0.678
15	Therm Cond Vap/Liq	W/m-C	0.000 / 0.133	0.000 / 0.133
16	Temperature In/Out	°C	-17.8 / 23.67	46.4 / -0.87
17	Operating Pressure (Abs)	kPa	60.000	60.000
18	Press. Drop Allow/Calc	kPa	10.000 / 13.174	10.000 / 12.803
19	Number of Passes/Shell		1	2
20	Velocity, Average	m/sec	0.26	0.47
21	Film Coef.	W/m²-C	905.48	660.86
22	Fouling Resist.	m²-C/W	0.000000	0.000000

23	Heat Duty	739 kW		MTD/Wtd/Corr		17.66 °C	F-CORR	0.898
24	Transfer Rate	150.15	Serv	323.45	Calc	323.45	Clean	0.00357 Foul

	Construction of One Shell					
25	TEMA Shell Type		E	Rear End Type		U.T.
26	Tube Type		PLAIN	Bundle Dia	mm	494
27	Tube O.D	mm	19.050	No. Holes/TubeSheet		250
28	Tube I.D	mm	14.834	No. Holes Counted		262
29	Area Ratio		1.284	Tube Pitch	mm	25.4000
30	Tube Length Total	m	6.09	Tube Layout Angle		30
31	Tube Length Effective	m	6.06	Impingement Plate		YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell		16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	mm	379
34	Window Area	m²	0.0310	In/Out Spacing	mm	379.0/379.0
35	Seal Strips		YES	Drop Under Noz In/Out	mm	33.9/33.9

	Shell Nozzles		**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia.	mm	101.60	101.60	Inside Dia.	mm	101.60	101.60
37	Velocity	m/sec	1.47	1.47	Velocity	m/sec	1.26	1.26
38	Rho-V-Sqr	kg/m-sec²	1631	1631	Rho-V-Sqr	kg/m-sec²	1267	1267
39	Nozzles/Shell	(OPP. SIDE)	1	1				

	Shellside Performance			**Pressure Drop**		
40	Bundle Flow Fraction		0.786	Shell Cross/Wind		5.494/3.335
41	Mass Vel Cross/Wind		192.3/289.9	Tubes		8.981
42	Mass Vel Long/Mean		72.5/236.1	Nozzles Shell/Tube		4.345/3.821
	Bundle Diameter Clearances			**Tube Metal Temperatures**		
43	Bundle-Shell	mm	5.08002	Avg. Tube Metal Temp.	°C	9.1
44	Baffle-Shell	mm	4.76250	Shellside Avg Surf. Temp	°C	9.0
45	Tube-Baffle	mm	0.92075	Tubeside Avg. Surf Temp	°C	9.3
46	Baffle Thk.	mm	6.350			

RESULTS OF DAY 97, TABLE A2 (ENGLISH UNITS)

HTC-STX	Version 3.5	Time: 4:10:37 PM		Date: 12/29/2001	File: BDFS_var	
*** Summary ***					English units	
Item No						
Service						
Calculation Mode	Rating Case					
Size	20 x 240	Type	AEU - HORZ	Connections 3 Series		1 Parallel
Surface/Unit	2,999		Shells/unit	3	Surf/Shell	999.69
Cost/Unit	42,466		Cost/Surf	14.16	Weight/Shell	5,390
Heat Duty	2,520,572		MTD	31.78	F-corr	0.8979
Rate-Service	26.44		Calculated	56.96	Calc Fouling	0.02026

	Shell	Tubes		
	Shell	**Tubes**	Tubes	0.750 x 0.083 on 1.00 30 deg
Flow Rate	71329	64852	Tube No	250 Type: PLAIN
Temperature In	-0.04	115.43	Baffles:	VERT SEG 14.9 space 25.0 cut
Temperature Out	74.6	30.4		
Pressure Drop	1.913	1.859	Surface Area	OK. Over design by 115.41%
Velocity	0.839	1.551	Shell pressure Drop	** Allowable exceeded.
Passes	1	2	Tube Pressure Drop	** Allowable exceeded.
Film Coef.	159.5	116.4	Vibration	OK. Within limits.
Nozzle In	1 x 4.0	4.0	Shell Nozzles	OK. Rho-V-Sqr within 4000
Nozzle Out	1 x 4.0	4.0	Chan Nozzles	OK. Rho-V-Sqr within 6000

RESULTS OF DAY 97, TABLE A2 (Cont.) (ENGLISH UNITS)

	HTC-STX	Version 3.5	Time: 4:10:37 PM		Date: 12/29/2001		File: BDFS_var
	*** Main ***						English units
1	Job No		Item No.		RATING Case		
2	Case Desciption						
3	TEMA Type	AEU - HORZ	Shell/Unit	3	Conn In	3 Series	1 Parallel
4	Size:	19.630 in Dia	239.8 in	Tube Length		in	Kettle Dia
5	Surface/Shell	ft²	1,003.9	Gross	999.7 Eff 23		U-Bend Area
6	Surface/Unit	ft²	3,011.7	Gross	2,999.1 Eff 69		U-Bend Area
	Performance of One Unit			**SHELLSIDE**			**TUBESIDE**
7	Fluid Circulated			shell side			tube side
8	Total Fluid In	lb/hr		71,329.2			64,851.6
9	Vapor	lb/hr		0.0			0.0
10	Liquid	lb/hr		71,329.2			64,851.6
11	Fluid Vap'z/Cond	lb/hr		0.0			0.0
12	Density In/Out	lb/ft³		46.946 / 46.946			49.942 / 49.942
13	Spec. Heat Vap/Liq	Btu/lb-F		0.000 / 0.490			0.000 / 0.443
14	Viscosity Vap/Liq	cP		0.000 / 0.546			0.000 / 0.678
15	Therm Cond Vap/Liq	Btu/hr-ft-F		0.000 / 0.077			0.000 / 0.077
16	Temperature In/Out	°F		0.0 / 74.61			115.4 / 30.43
17	Operating Pressure (Abs)	psi		8.702			8.702
18	Press. Drop Allow/Calc	psi		1.450 / 1.913			1.450 / 1.859
19	Number of Passes/Shell			1			2
20	Velocity, Average	ft/sec		0.84			1.55
21	Film Coef.	Btu/hr-ft²-F		159.47			116.39
22	Fouling Resist.	hr-ft²-F/Btu		0.000000			0.000000
23	Heat Duty	2,520,572 Btu/hr	MTD/Wtd/Corr		31.78 °F	F-CORR	0.898
24	Transfer Rate	26.44	Serv	56.96 Calc	56.96	Clean	0.02026 Foul
	Construction of One Shell						
25	TEMA Shell Type		E	Rear End Type			U.T.
26	Tube Type		PLAIN	Bundle Dia	in		19.43
27	Tube O.D	in	0.750	No. Holes/TubeSheet			250
28	Tube I.D	in	0.584	No. Holes Counted			262
29	Area Ratio		1.284	Tube Pitch	in		1.0000
30	Tube Length Total	ft	19.98	Tube Layout Angle			30
31	Tube Length Effective	ft	19.89	Impingement Plate			YES
32	Baffle Type		VERT- SEG	Crosspasses/Shell			16
33	Baffle Cut, Frac Dia/NFA		0.340/0.250	Central Spacing	in		14.921
34	Window Area	in²	48.0492	In/Out Spacing	in		14.9/14.9
35	Seal Strips		YES	Drop Under Noz In/Out	in		1.3/1.3
	Shell Nozzles	**Inlet**	**Outlet**	**Tube Nozzles**		**Inlet**	**Outlet**
36	Inside Dia. in	4.00	4.00	Inside Dia.	in	4.00	4.00
37	Velocity ft/sec	4.84	4.84	Velocity	ft/sec	4.13	4.13
38	Rho-V-Sqr lb/ft-sec²	1098	1098	Rho-V-Sqr	lb/ft-sec²	853	853
39	Nozzles/Shell (OPP. SIDE)	1	1				
	Shellside Performance			**Pressure Drop**			
40	Bundle Flow Fraction		0.786	Shell Cross/Wind			0.798/0.484
41	Mass Vel Cross/Wind		39.4/59.4	Tubes			1.304
42	Mass Vel Long/Mean		14.8/48.4	Nozzles Shell/Tube			0.631/0.555
	Bundle Diameter Clearances			**Tube Metal Temperatures**			
43	Bundle-Shell	in	0.20000	Avg. Tube Metal Temp.	°F		48.4
44	Baffle-Shell	in	0.18750	Shellside Avg Surf. Temp	°F		48.2
45	Tube-Baffle	in	0.03625	Tubeside Avg. Surf Temp	°F		48.7
46	Baffle Thk.	in	0.250				

Appendix D
Regression Analysis

Regression Analysis

In many problems there are several variables that are related by certain functional relationship and it is of interest to model and explore this relationship. In general, suppose that there is a dependent variable or response y (such as correction factors) that depends on several independent or regressor variables x_1, x_2, …., etc (such as tube OD, velocity, pressure, temperature, etc). The response y could be written as:

$y = B_0 + B_1 X_1 + B_2 X_2 + B_3 X_3 + \ldots\ldots + B_k X_k + \varepsilon$

Where:

B_j , $j = 0, 1, \ldots, k$ are called the regression coefficients

$X_1 = f_1(x_1, x_2, \ldots, x_k)$

$X_2 = f_2(x_1, x_2, \ldots, x_k)$

….

$X_k = f_k(x_1, x_2, \ldots, x_k)$

ε is the error or residual

k is the number of regression coefficients

Using the matrix notation for:

$$
y = \begin{pmatrix} y_1 \\ y_2 \\ .. \\ .. \\ y_n \end{pmatrix}
$$

$$
X = \begin{pmatrix} 1 & X_{11} & X_{12} & \ldots\ldots & X_{1k} \\ 1 & X_{21} & X_{22} & \ldots\ldots & X_{2k} \\ \ldots & \ldots & \ldots & \ldots & \ldots \\ 1 & X_{1k} & X_{2k} & \ldots\ldots & X_{nk} \end{pmatrix}
$$

$$
B = \begin{pmatrix} B_1 \\ B_2 \\ .. \\ B_k \end{pmatrix}
$$

Where n is the number of experimental data.

It could be shown using least squares estimators that:

$B = (X' X)^{-1} X' y$

Where X' denotes the transport of X

The error or residual sum of squares SS_E is shown to be:

$SS_E = y' y - B' X' y$

The regression sum of squares SS_R will be:

$SS_R = B' X' y - (\Sigma y_i)^2 / n$

And the total sum of squares will be:

$SST = y' y - (\Sigma y_i)^2 / n$

The coefficient of determination R^2, which is a measure of how well the model represents the experimental data, will be:

$R^2 = SS_E / SS_T$

The significant of the model coefficients B_j could be tested using the t test. The coefficient under consideration is significant if $|t_o| > t_{\alpha/2,\, n-k-1}$

Where:

$t_o = B_j / (MS_E\, C_{jj})^{0.5}$

$MS_E = SS_E / (n-k-1)$

α is the level of significance

and C_{jj} is the diagonal elements of $(X' X)^{-1}$ corresponding to B_j.

NOMENCLATURE

A	:	heat transfer area, m^2
Ar	:	Archimedes number
B_c	:	baffle cut, Degree
C_p	:	specific heat, J/kg K
D_i	:	tube inside diameter, m
D_o	:	tube outside diameter, m
D_{OTL}	:	bundle diameter, m
d_p	:	particle diameter, m
D_S	:	shell internal diameter, m
e	:	porosity, dimensionless
f	:	friction factor, dimensionless
F	:	temperature correction factor, dimensionless
F_{bp}	:	bypass area fraction, m^2
F_c	:	fractions of the tubes in cross flow, dimensionless
F_{cr}	:	fraction of cross flow, dimensionless
F(t)	:	cumulative distribution function, dimensionless
g	:	gravitational acceleration, 9.81 m^2/s
h	:	heat transfer coefficient, W/m^2 K

h_c	:	cross flow heat transfer coefficient, W/m^2 K
h_{lv}	:	latent heat of evaporation, J/kg
H(t)	:	cumulative hazard function, dimensionless
I	:	irreversibility, W
J_B	:	bypass correction factor for heat transfer coefficient, dimensionless
J_c	:	correction factor for baffle configuration for heat transfer coefficient, dimensionless
J_L	:	leakage correction factor for heat transfer coefficient, dimensionless
k	:	thermal conductivity, W/m K
K_f	:	Euler number, dimensionless
K_{sc}	:	pressure drop correction factor, dimensionless
L	:	tube length, m
L_B	:	baffle spacing, m
L_c	:	baffle cut distance, m
L_s	:	shell length, m
$\dot{m}$	:	mass flow rate, kg/s
n	:	number of tube side passes
N	:	number of baffles
N_f	:	number of failure (time to reach to the critical level of fouling)
N_c	:	number of cross flow rows
N_{cw}	:	effective number of cross flow rows in the windows zone
N_T	:	number of tubes

NTU : number of transfer unit, dimensionless

Nu : Nusselt number, dimensionless

P : thermal effectiveness, dimensionless

P^* : modified thermal effectiveness, dimensionless

Pr : Prandtle number, dimensionless

P_T : tube pitch, m

$\dot{Q}$: heat transfer rate, W

R : heat capacity ratio, dimensionless

R(t) : reliability function, dimensionless

R^2 : coefficient of determination, dimensionless

R_B : bypass correction factor for pressure drop, dimensionless

Re : Reynolds number, dimensionless

Re_{sc} : Modified Reynolds number of self cleaning heat exchanger, dimensionless

R_k : conduction resistance $\frac{D_o}{2k}\ln\left(\frac{D_o}{D_i}\right)$, m^2 K/W

R_L : leakage correction factor for pressure drop, dimensionless

R_f : fouling resistance, m^2 K/W or hr ft^2 F/Btu

s : entropy, J/kg K

S_b : bypass flow area, m^2

S_m : window zone flow area, m^2

S_s : Shell to baffle leakage area for FS method, m^2

S_{sb} : shell to baffle leakage area for BD method, m^2

S_t : tube to baffle leakage area for FS method, m^2

S_{tb} : tube to baffle leakage area for BD method, m^2

S_w : window zone flow area, m^2

$\dot{S}_{gen}$: entropy generation, W/K

t : time, sec

T : temperature, C

t_b : thickness of baffle, m

T_{cond} : condensation temperature, C

T_{evap} : evaporation temperature, C

T_o : ambient temperature, C

T_{sur} : surface temperature, C

U_a : actual overall heat transfer coefficient, W/m^2 K

U_c : Theoretical (clean) overall heat transfer coefficient, W/m^2 K

V : velocity, m/s

$V_{1,s}$: superficial velocity, m/s

V_{max} : maximum intertube velocity, m/s

V_t : tube side velocity, m/s

GREEK SYMBOLS

ε : effectiveness, dimensionless

β : power of the fouling power law model, dimensionless

ρ : density, kg/m^3

σ^2 : variance of the distribution, dimensionless

μ : viscosity, $N\ s/m^2$

μm : mean of the distribution, dimentionless

ΔT : temperature difference, C

ΔT_{ln} : log mean temperature difference, C

ΔP : pressure drop, Pa

ΔP_c : cross flow pressure drop, Pa

ΔP_w : window zone pressure drop, Pa

Δ_b : bundle to shell diametral clearance, m

Δ_{sb} : shell to baffle diametral clearance, m

Δ_{tb} : tube to baffle diametral clearance, m

SUBSCRIPT

c : cold side

cond : condensation

evap : evaporation

h : hot side

in : inlet

l : liquid

out	:	outlet
pc	:	phase change
s	:	shell side
sat	:	saturation
t	:	tube side
tot	:	total
v	:	vapor
wtd	:	weighted

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EDUCATION

1998-Data : King Fahd University of Petroleum and Minerals (KFUPM), Mechanical Engineering Department, Dhahran, Saudi Arabia.

Expected to graduate by May 2002 with a PhD degree in Mechanical Engineering. I took courses related to boundary elements, probabilistic concepts, computational methods on ChE, applied regression, radiation heat transfer, turbulence, advanced thermodynamics, elasticity, combustion and emission, and heat exchangers design.

My PhD Dissertation was related to heat exchangers. My dessertation title is "Fouling Analysis and its Mitigation in Heat Exchangers ". In this dessertation thermal, 2nd Law and statistical analyses were performed to two exchangers. Also, the effectiveness of two new fouling mitigation technologies was evaluated.

Having excellent knowledge in numerical work, which involve finite difference, finite element, boundary element and collocation method.

1992-1995 : King Fahd University of Petroleum and Minerals (KFUPM), Mechanical Engineering Department, Dhahran, Saudi Arabia.

Graduate with Master Degree (January 1995) in Mechanical Engineering with a GPA 4.00 on 4.00 scale. I took courses related to power plants, advanced fluid mechanics, heat conduction, heat convection, numerical solution to PEDs, continuum mechanics and advanced engineering mathematics. My thesis title is "Energy and Exergy Analysis of Ghazlan Power Plant".

1987-1992 : King Fahd University of Petroleum and Minerals (KFUPM), Mechanical Engineering Department, Dhahran, Saudi Arabia.

Graduate with Bachelor Degree (January 1992) in Mechanical Engineering with a GPA 3.34 on 4.00 scale. I took courses related to energy conversion, compressible fluid flow, fluid power systems, mechanical vibrations and system dynamics and control. I am familiar with machine design and design concepts. My senior project was related to energy evaluation of gas and steam turbines.

WORK EXPERIENCE

1995-Data : SAUDI ARAMCO, Dhahran, Saudi Arabia.

Working with Consulting Services Department as a Heat Exchangers engineer. Reviewed several projects and recommended several modifications to several exchangers in many plants. These recommendations resulted in considerable savings (millions of Riyals per year).

Worked in Abqaiq Plants and Ras Tanura Refinery as operation engineer.